AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 43 No. 563 FEBRUARY, 1953 PRICE: 3s. 6d.

E S C Torsion Bars are included as standard equipment in many makes of cars. They are also fitted in a large number of cars made for racing and competition work.

A tribute to their sterling quality comes from one of the leading makers who states—"Throughout the many races and trials in which our cars have participated, often with considerable success, not once have your Torsion Bars given the slightest sign of trouble".

ENGLISH STEEL CORPORATION LTD.

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nothing rattles the NYLOC

Sh-Sh-Shake it how you will, the great thing about the Nyloc Nut is that, once it is screwed on, nothing can budge it except a spanner. Its special nylon insert locks so tightly on the bolt that you can vibrate it for hours on end and it still won't loosen. Heat it up to 200°C, freeze it to -65°C, soak it in oil or moisture — the Nyloc always stays put and it can be used over and over again.

The Fibre Nut with its fibre insert and the Pinnacle Nut with its metal diaphragm are equally tenacious. If you have any problem that involves vibration, one of these three is the nut you are looking for. Let us advise you which is best for your particular job.

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SIMMONDS AEROCESSORIES LIMITED · TREFOREST · GLAMORGAN

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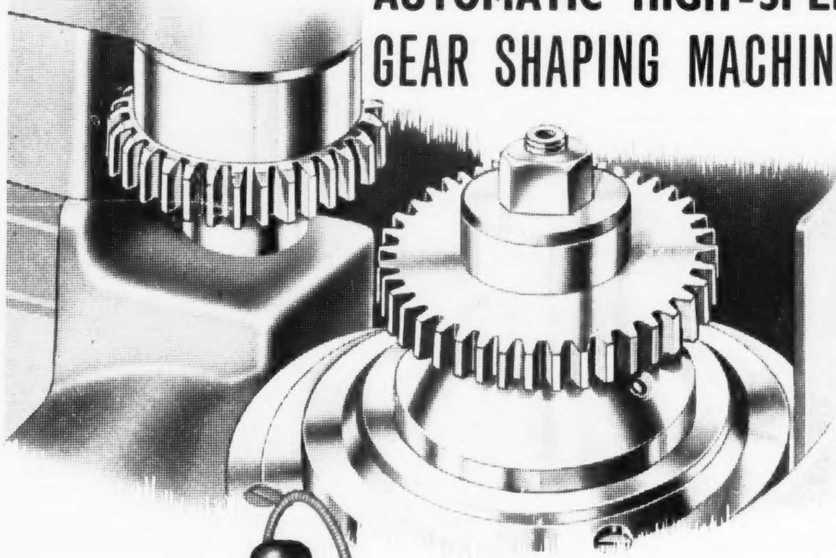
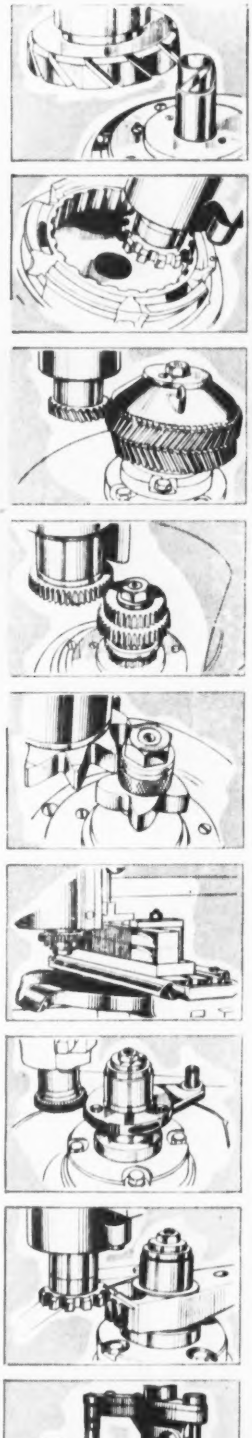
LORENZ

AUTOMATIC HIGH-SPEED GEAR SHAPING MACHINES

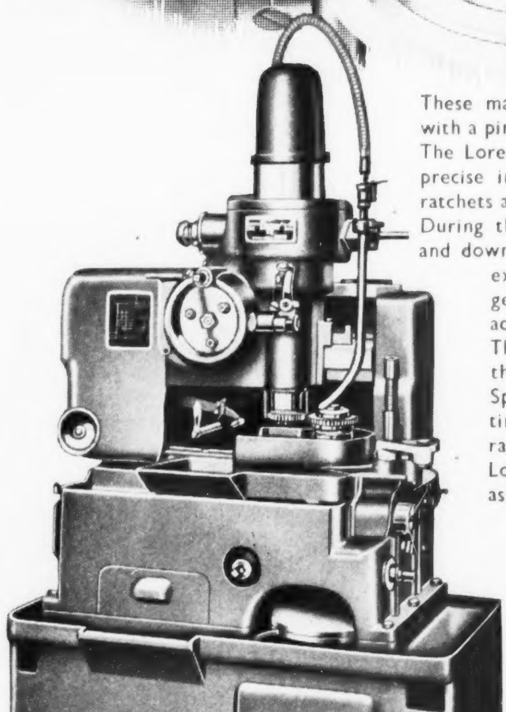




These machines shape by the generating process with a pinion type cutter for spur and helical gears. The Lorenz method is ideal for the production of precise internal and external gearing, sprockets, ratchets and other profiles.

During the cutting stroke the tool is moved up and down and rotated so that the tool and work execute the movement of two meshing gears. This method ensures maximum accuracy and smooth flanks of the work. The tool is re-sharpened by simply grinding the tooth face.

Special attachments are available for generating helical gears, gear trains, face gears, rack gears, shaping between centres, etc. Lorenz gear shapers are made in five sizes as shown below.



Model SV00

AUTOMATIC GEAR SHAPERS

MODEL	S 00	SV 00	S 5	S7/500	S7/1000
Pitch diam.	7⅛"	7⅛"	17¾"	19⅝"	40"
Face width	2"	2"	3"	5⅛"	6.5/16"
Diam. pitch/module	6/4	6/4	5/5	3,6/7	3,6/7

AUTOMATIC GEAR HOBBERS

High production machines of extreme accuracy for the manufacture of spur, helical, worm wheels, and special gearings are made for work of pitch diameters from 23½" to 63" and diam. pitch/module 4.23/6 to 1.59/16.

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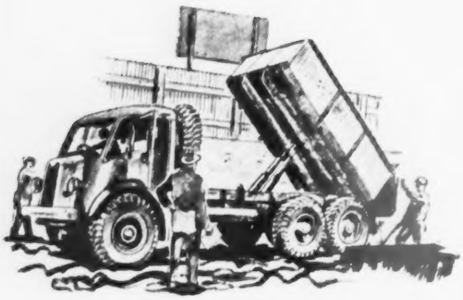

Dunlop Flexible Pipes put an end to pipe difficulties. Consider some of the practical advantages which they offer: durability and high fatigue-resistance, resulting from special Dunlop design and the traditional quality of materials and workmanship; short end-fittings that offer ease of installation; secure end-fittings that cannot be shifted by internal pressure. There are many types of coupling for original fitment or replacement, and, of course, high, medium, and low pressure ranges. The Dunlop Technical and Consultative Service is ready to advise you on all application problems.

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The new
'STRAP DRIVE'
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25 YEARS
SPECIALISATION
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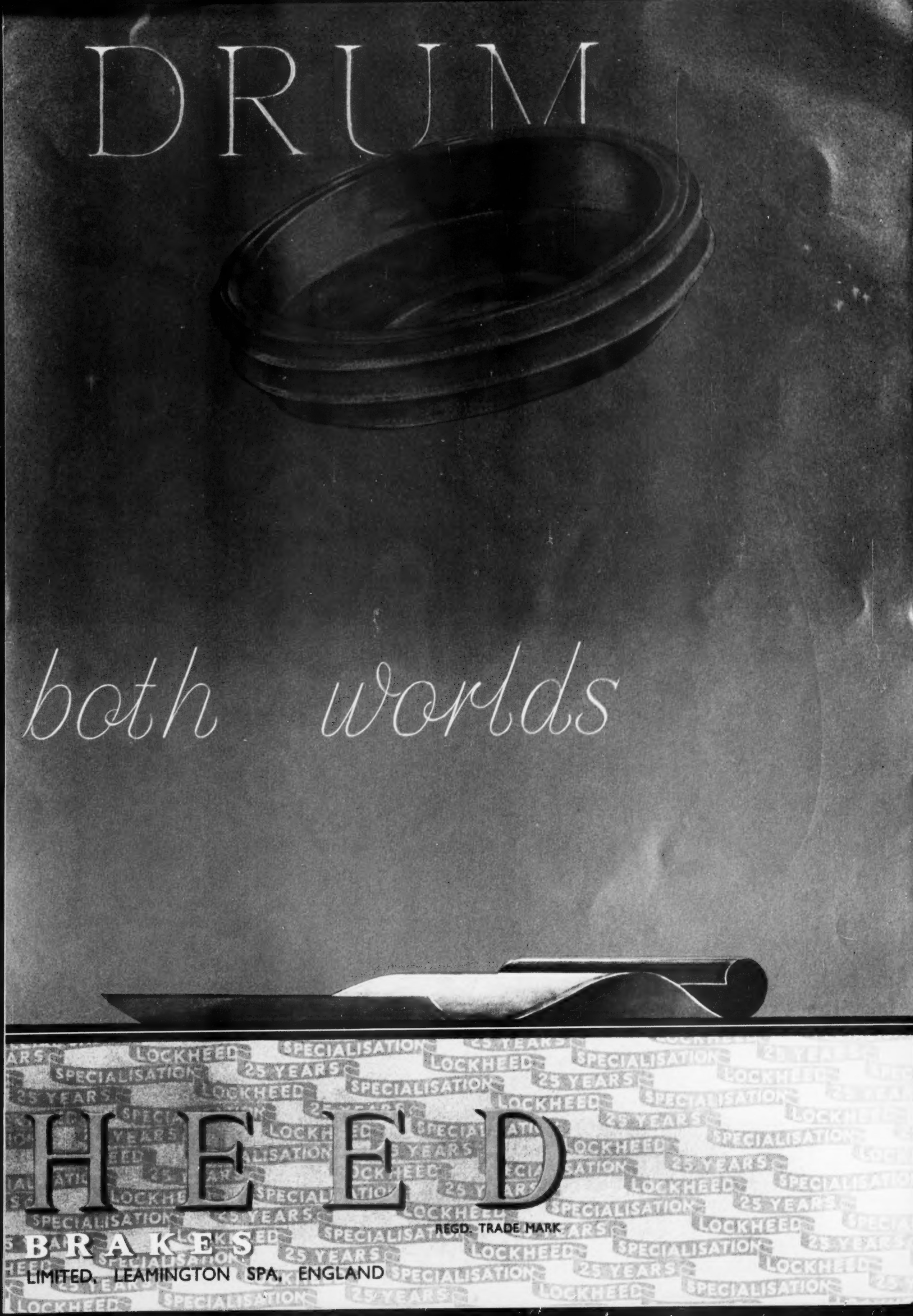
HYDRAULIC
AUTOMOTIVE PRODUCTS COMPANY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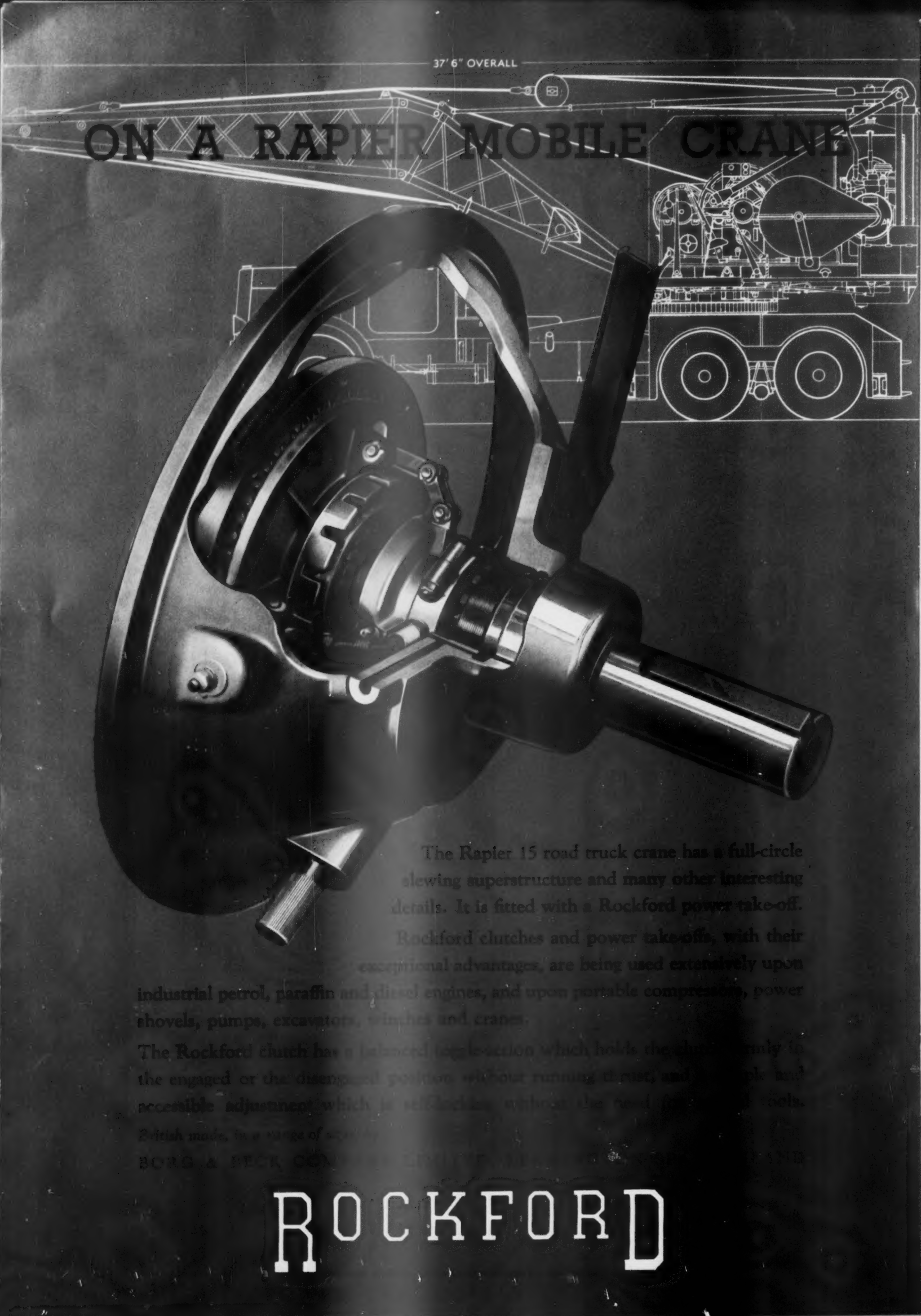
37' 6" OVERALL
ON A RAPIER MOBILE CRANE
The Rapier 15 road truck crane has a full-circle slewing superstructure and many other interesting details. It is fitted with a Rockford power take-off.
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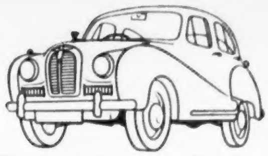
The Rockford clutch has a balanced toggle-action which holds the
the engaged or the disengaged position without running thrust, and
accessible adjustment which is self-locking, without the need
British made, in a range of
BORG & BECK
ROCKFORD

THINK ZINC



Almost as quickly as you can say the words "Think Zinc", molten zinc alloy has become the Austin A40 radiator grille assembly shown here. This speed in production is just one reason why the automobile industry — like so many other industries — has adopted the zinc alloy die casting process. Further advantages are: —

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STRENGTH · LONG LIFE OF DIES · LITTLE NEED FOR MACHINING · LOW UNIT COST

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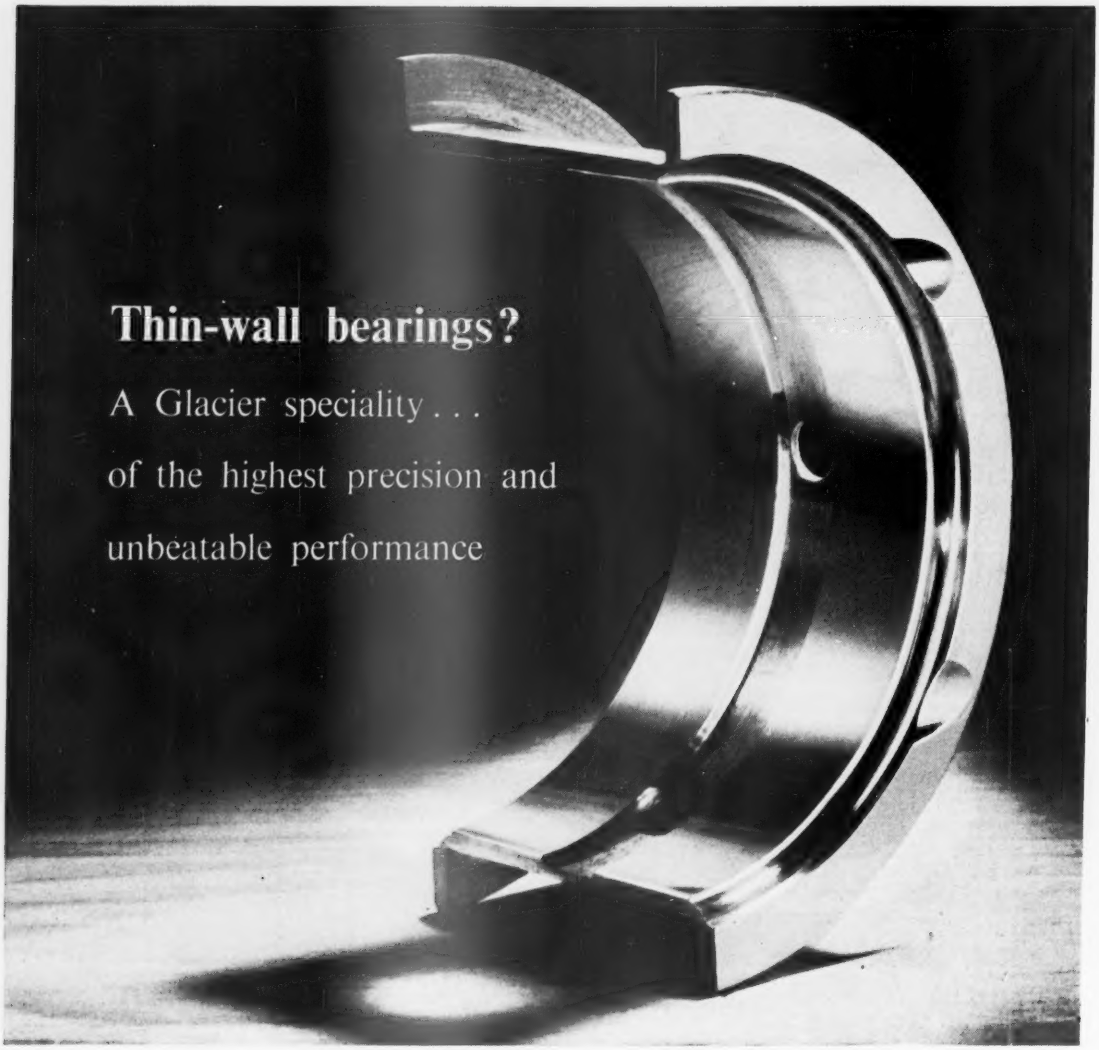

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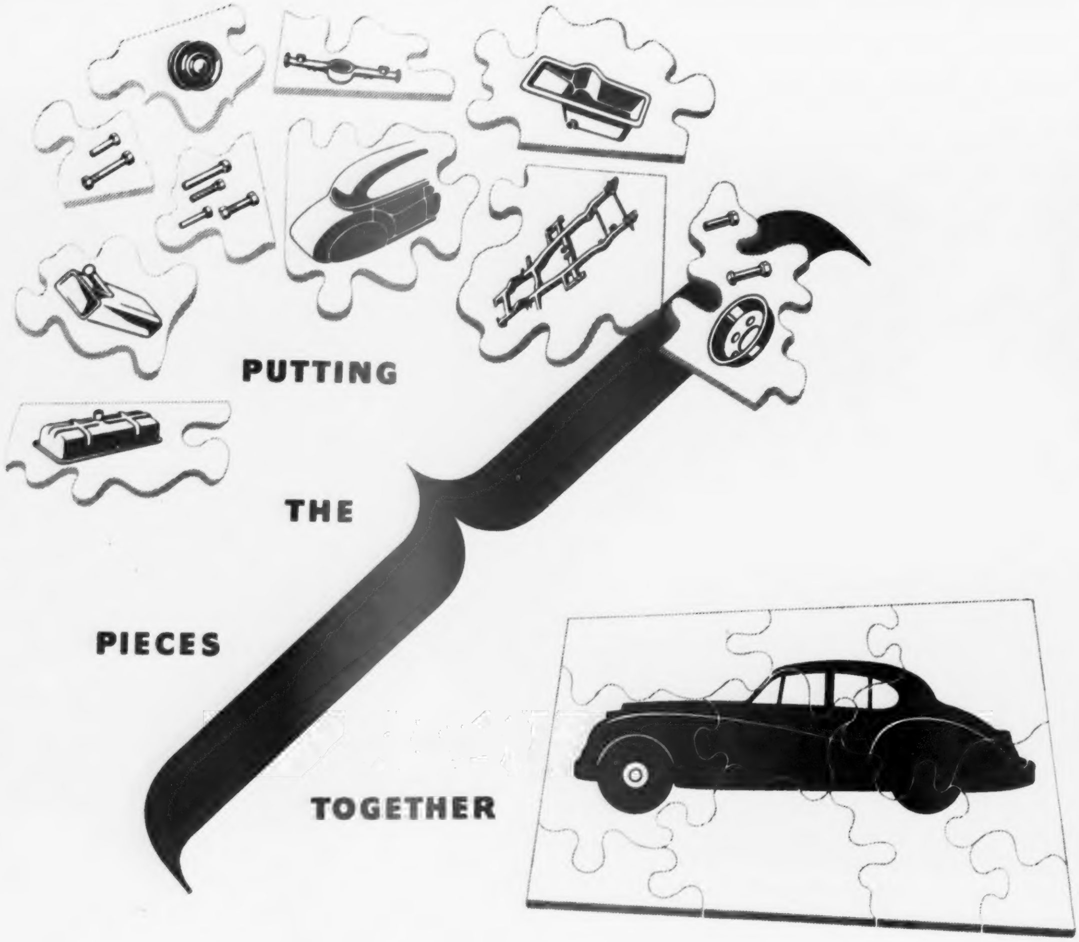
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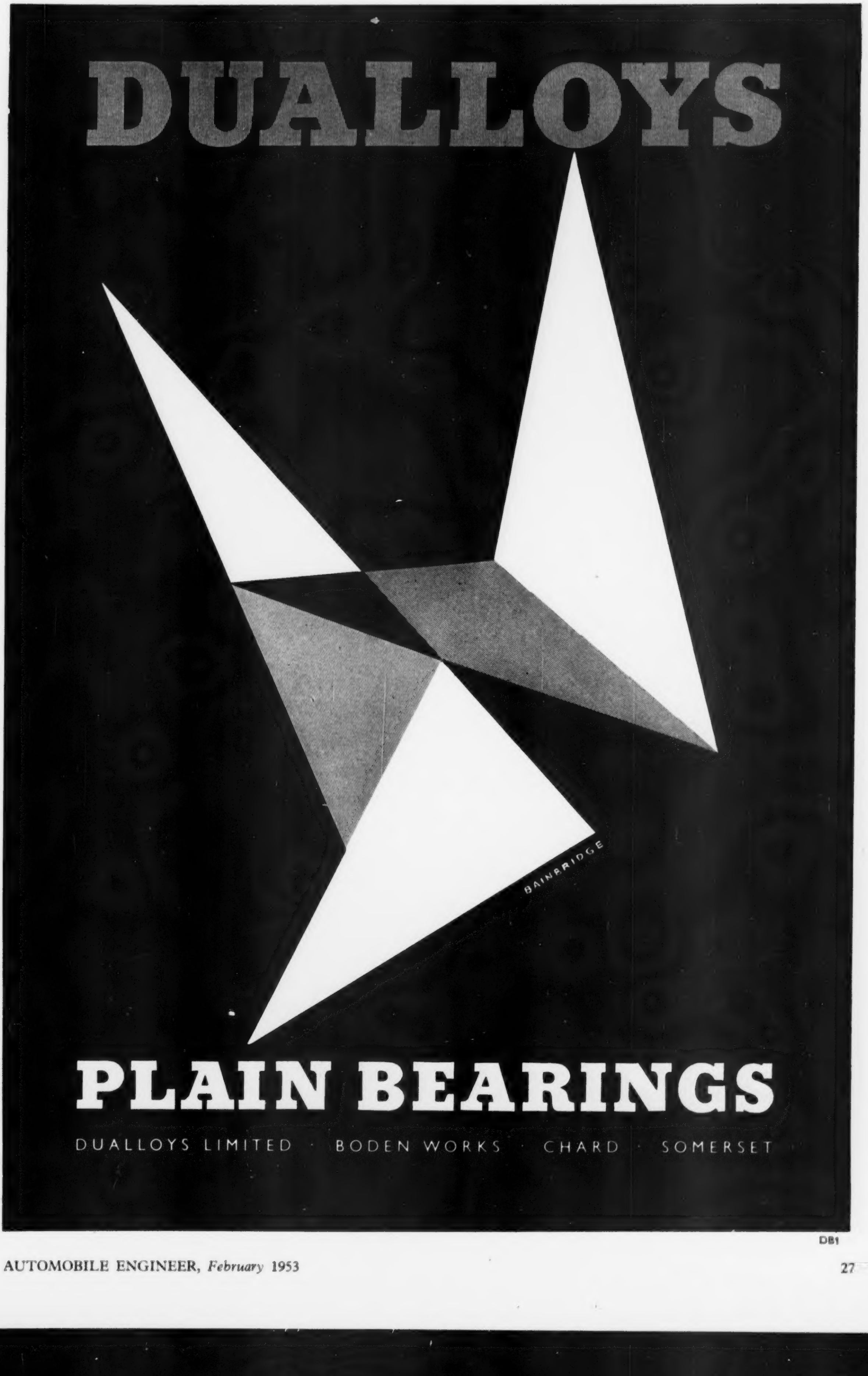
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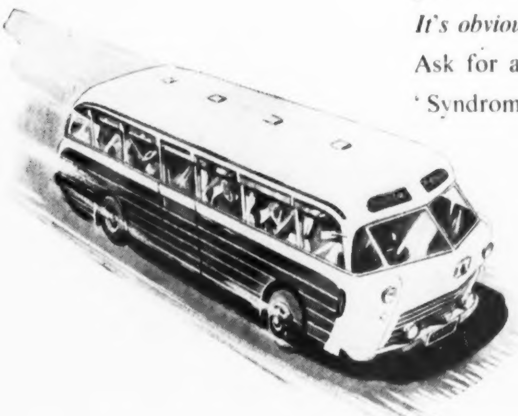
Ask for a Tecalemit technical representative to call and discuss the 'Syndromic' system with you.



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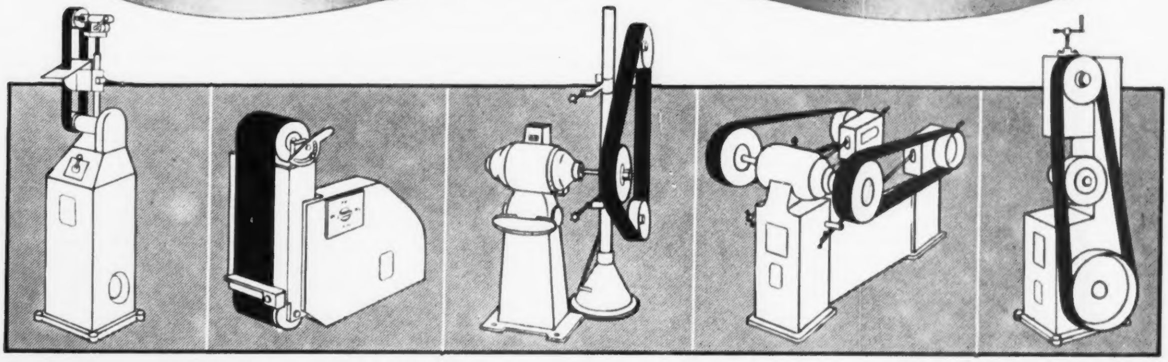

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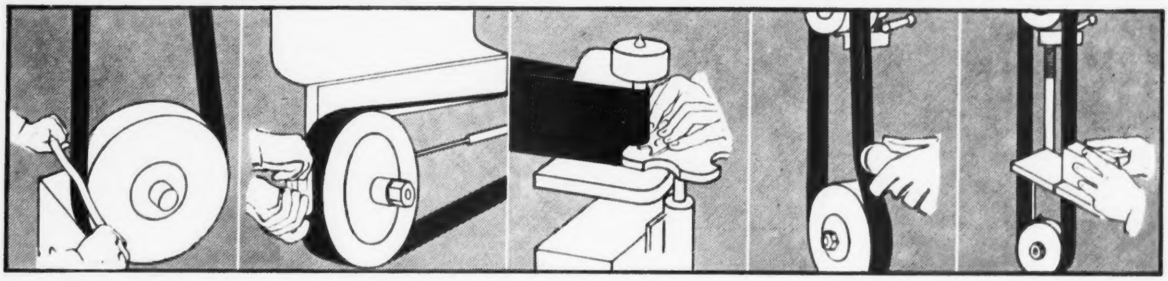

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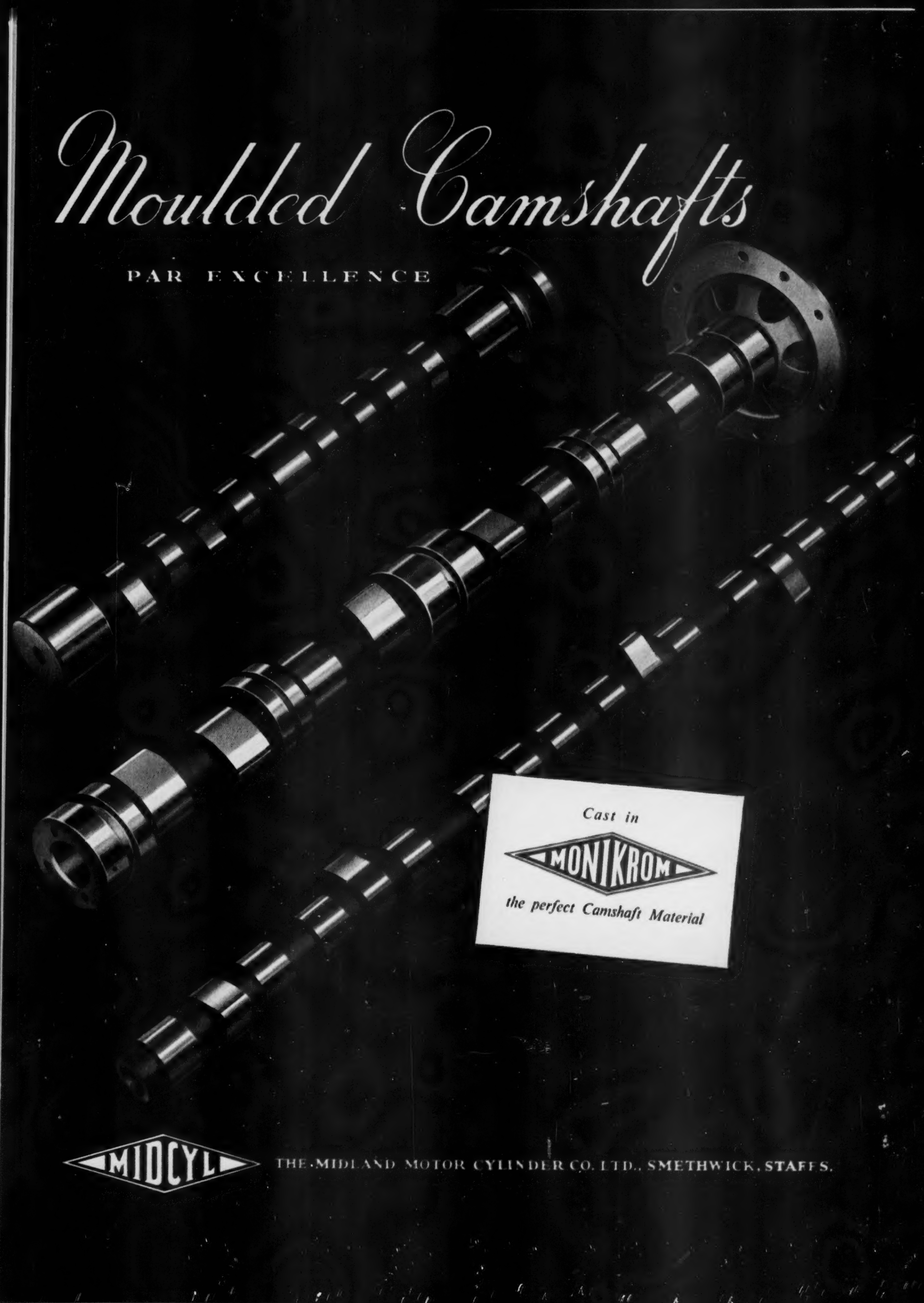
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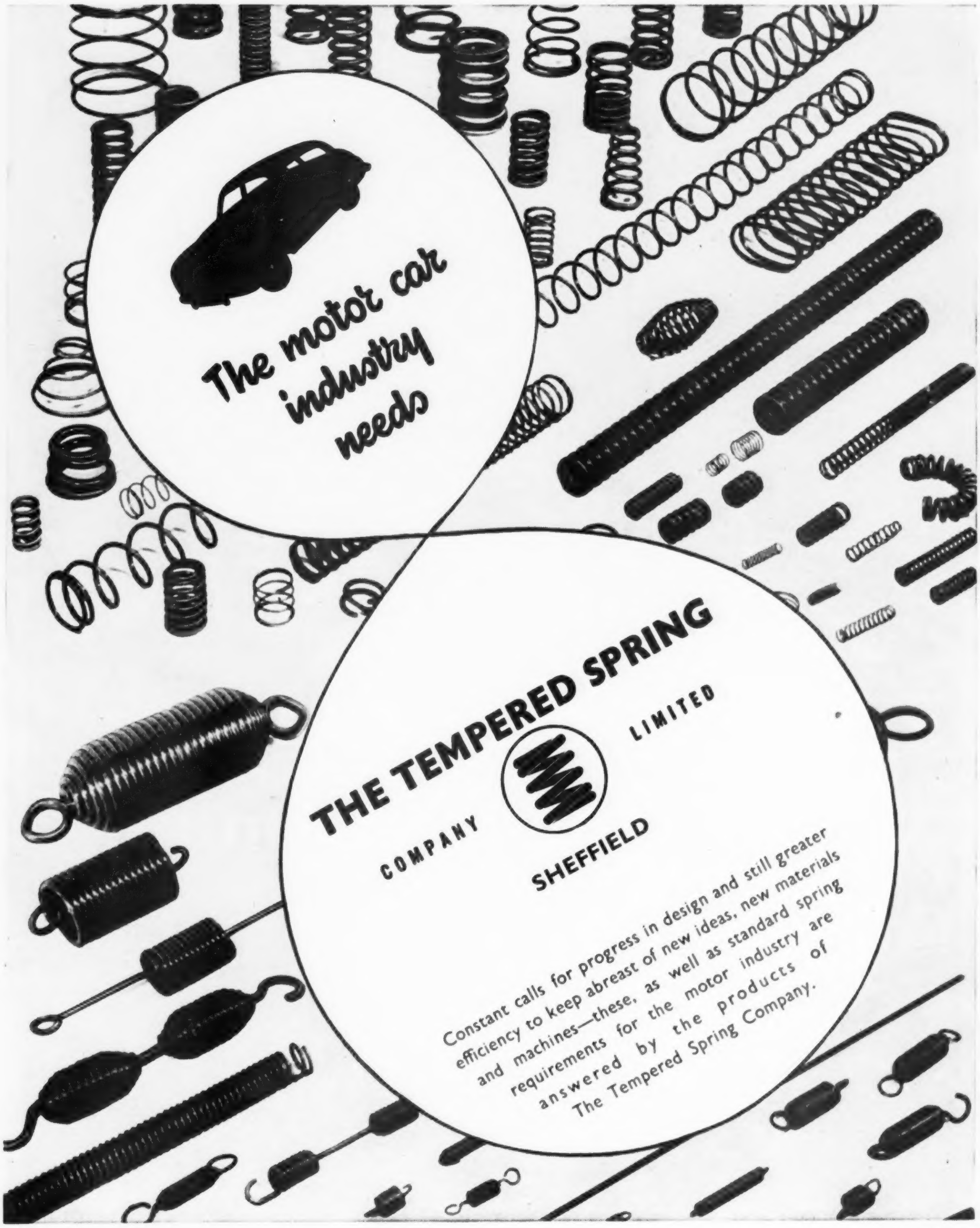
THE MIDLAND MOTOR CYLINDER CO. LTD., SMETHWICK, STAFFS.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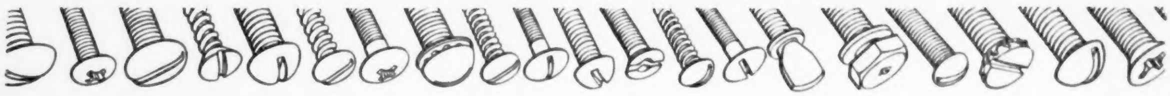

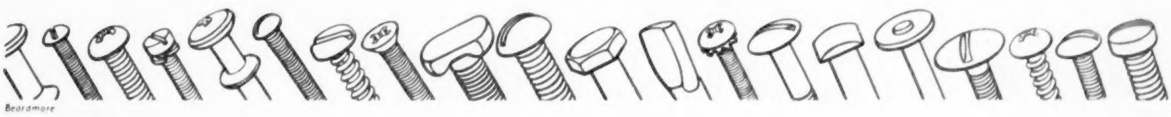

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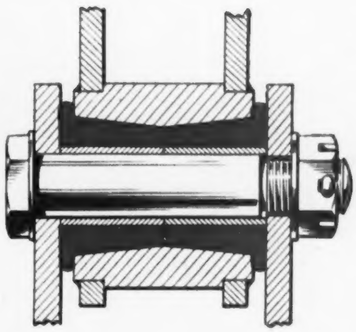

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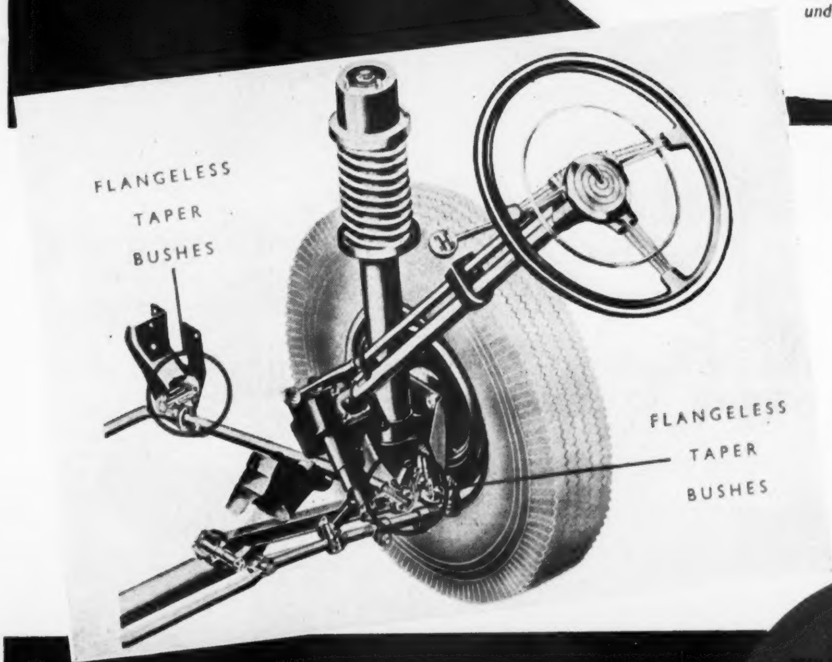
Right: Flangeless Taper Bush sectioned to show construction. In the housing flanges develop under compression.



★After exhaustive testing Ford Motor Company Ltd. now fit Silentbloc Flangeless Taper Bushes in the I.F.S. of the "Consul" and "Zephyr Six".

Left: Front Suspension of the "Consul" and "Zephyr Six".

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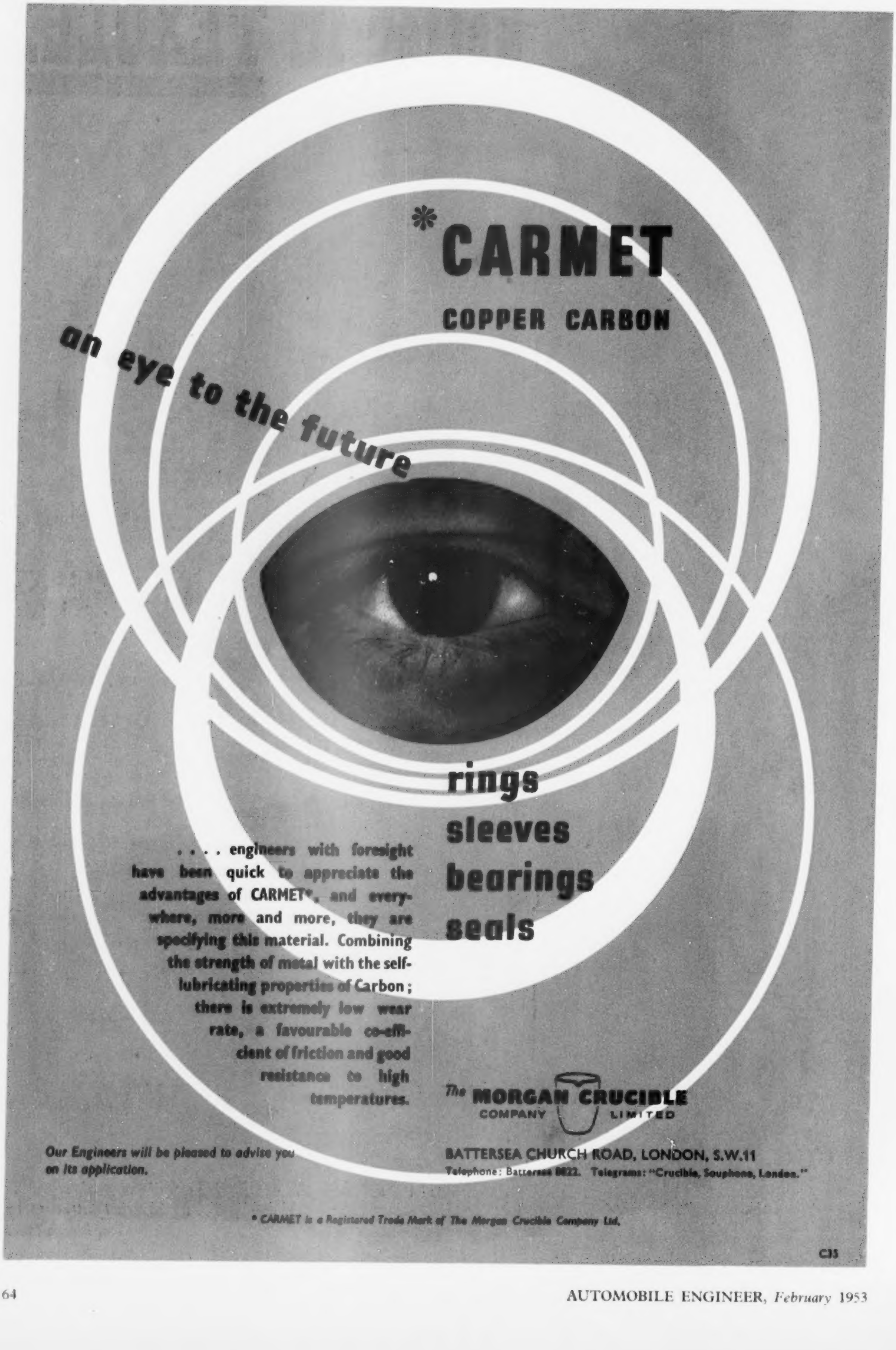
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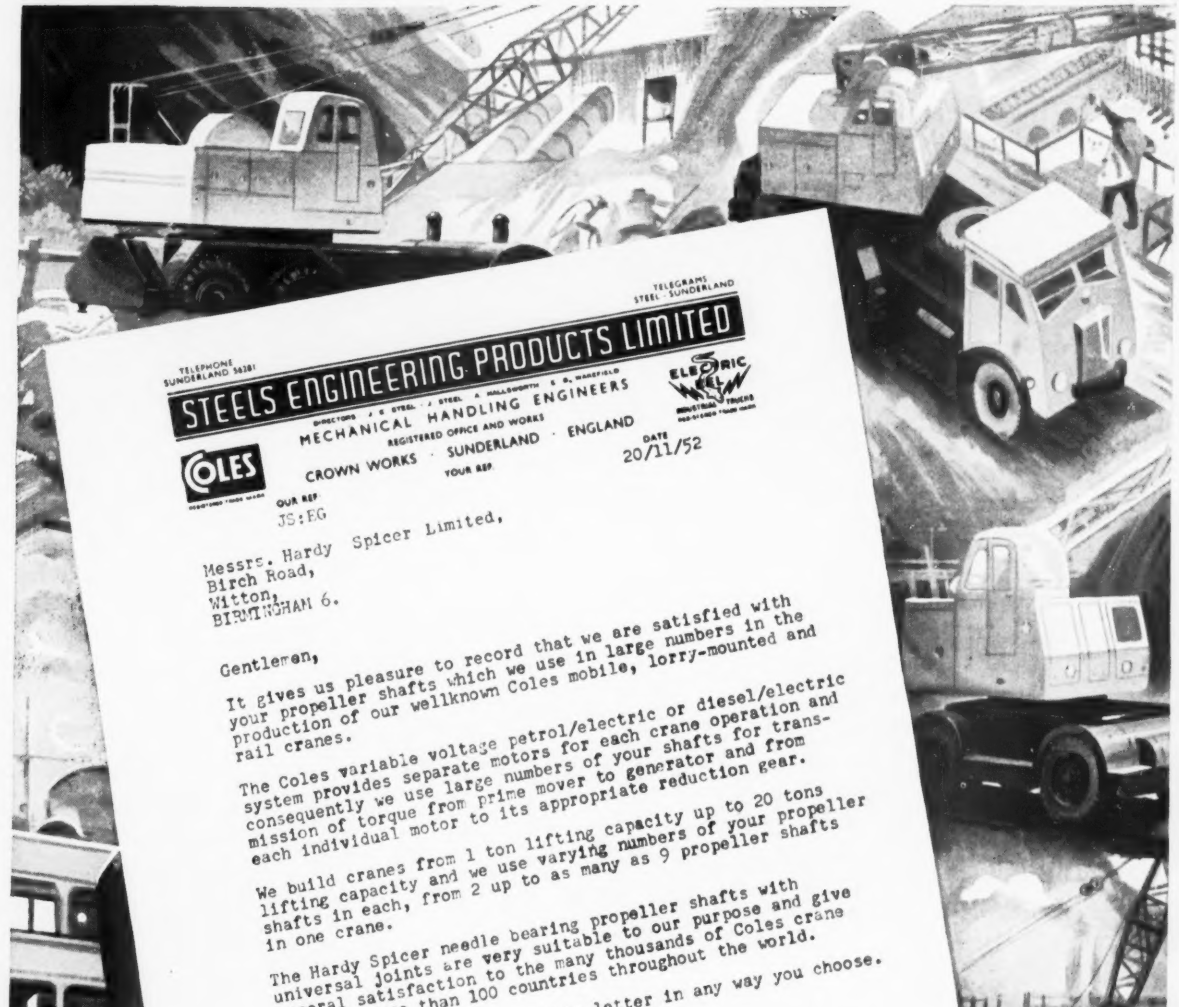

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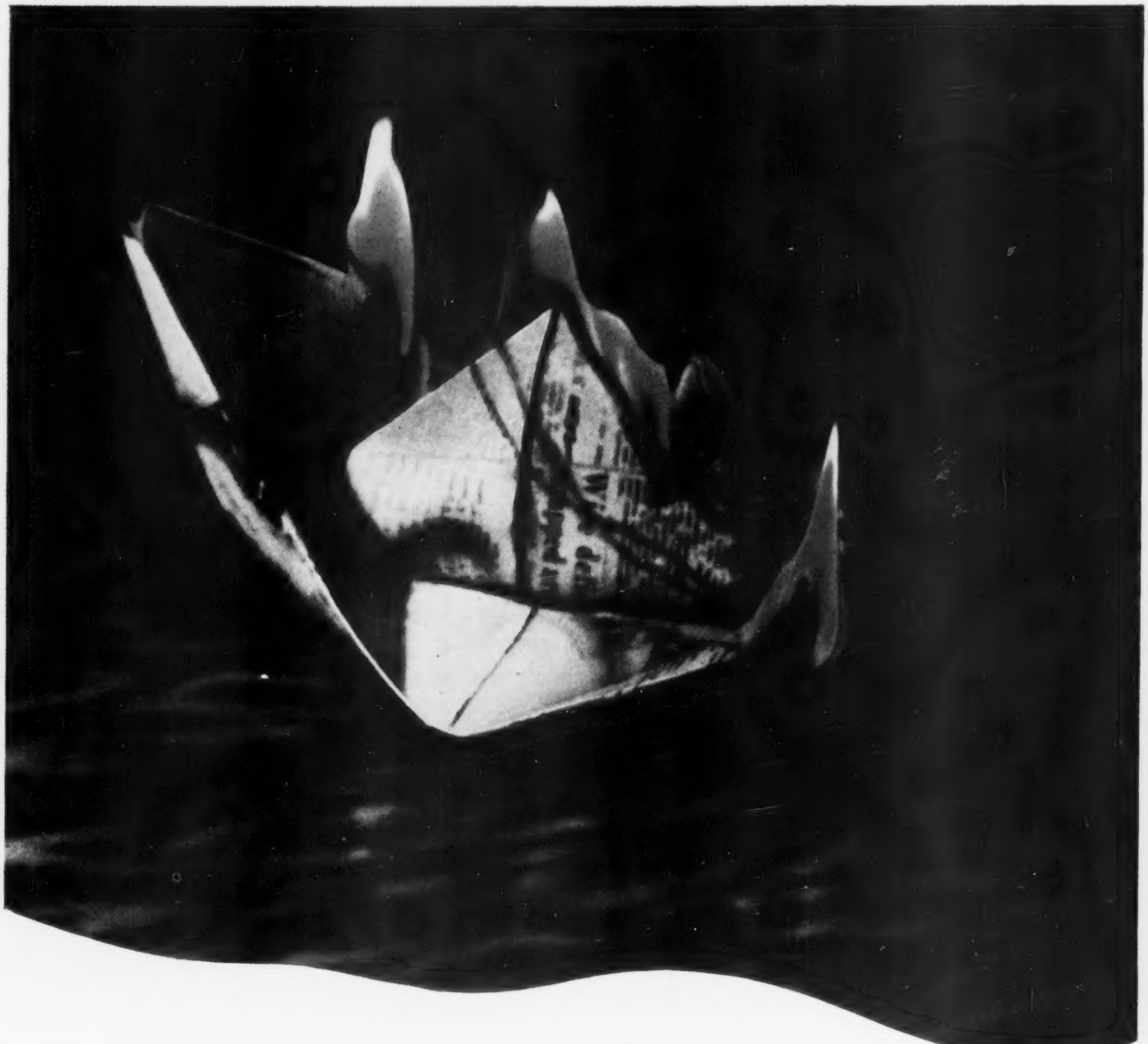

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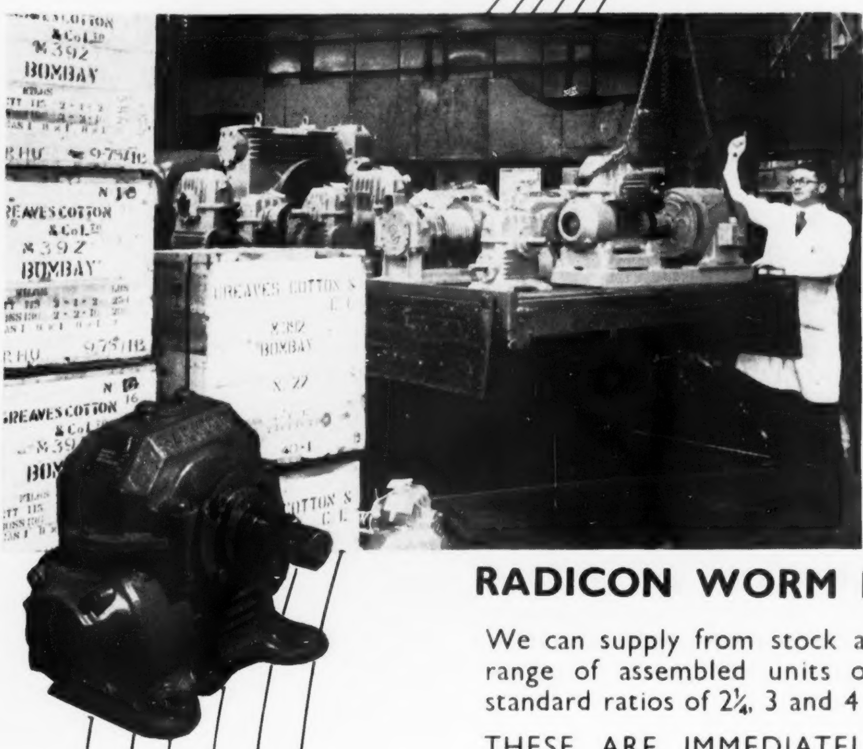

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THESE ARE IMMEDIATELY AVAILABLE

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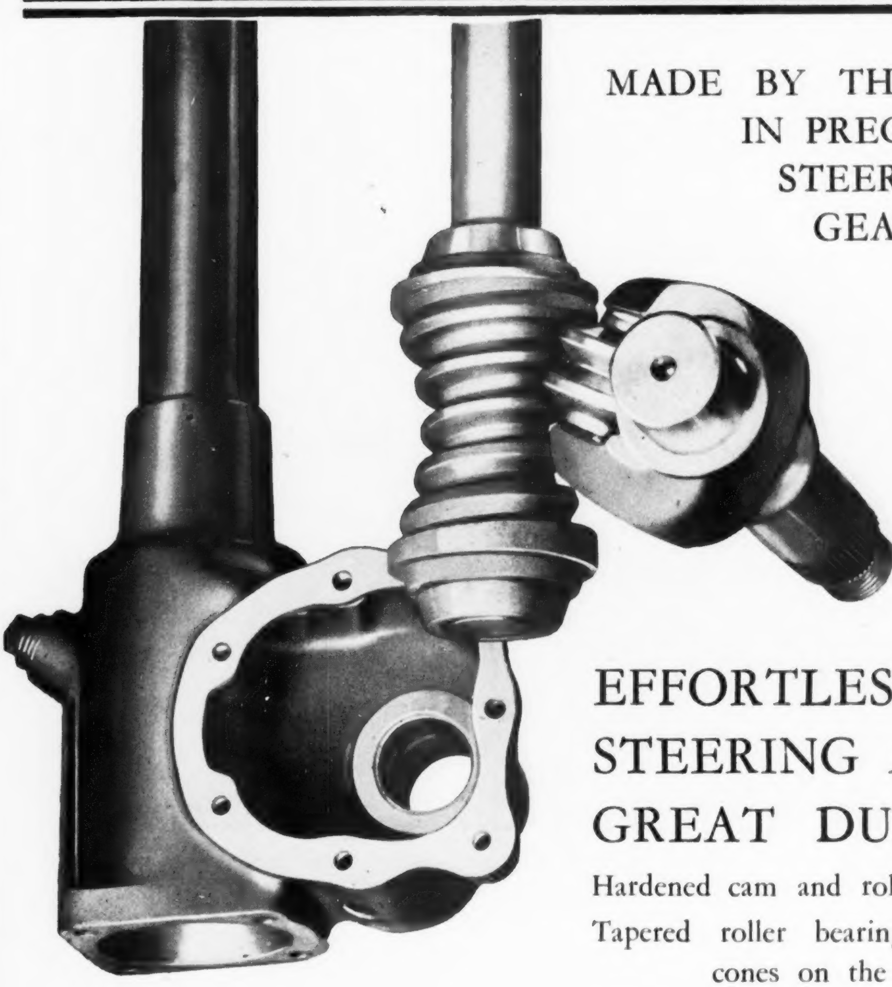

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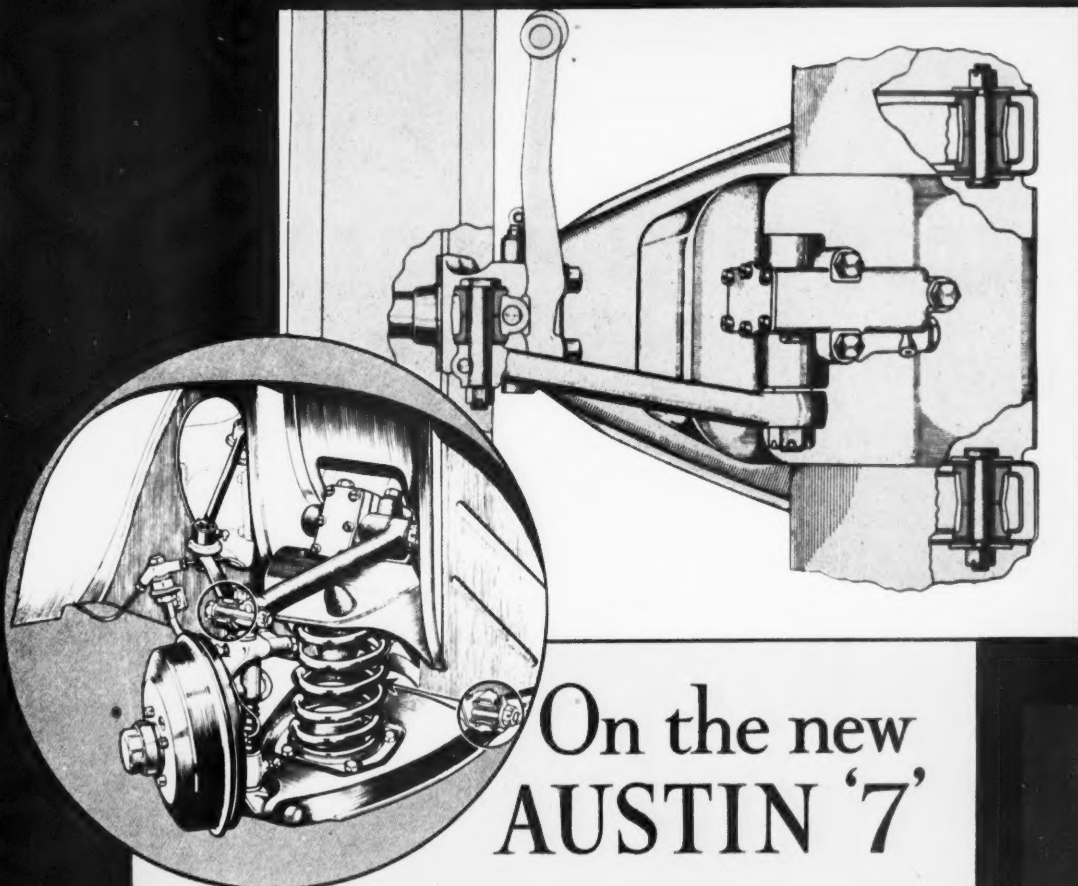

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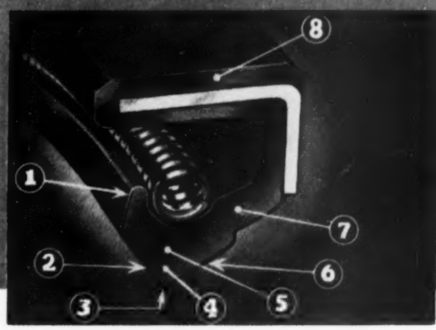

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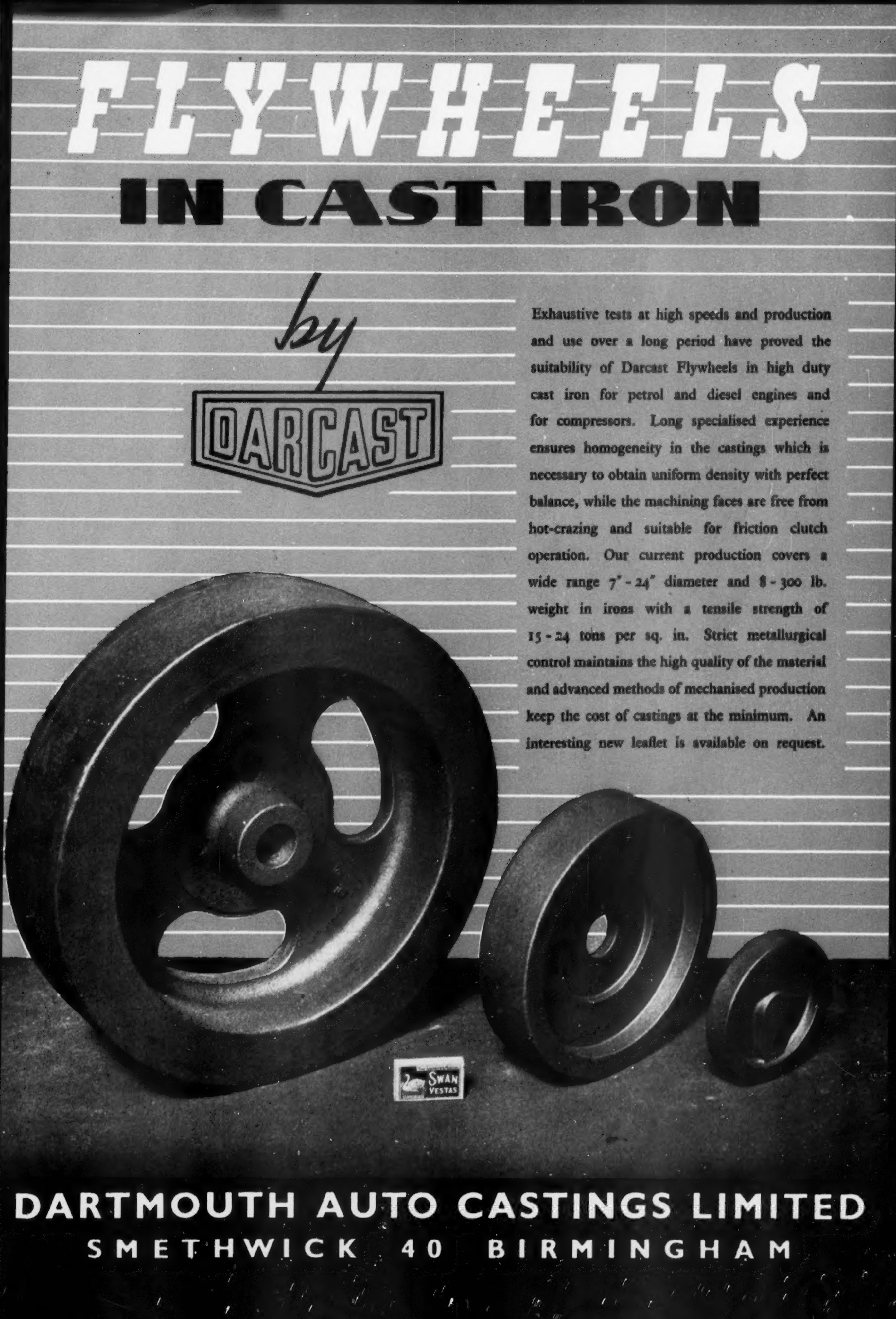
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Design, Materials, Production Methods, and Works Equipment

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Low Viscosity Oils

THAT advantages may be obtained from the use of low viscosity oils has been recognized for many years. In the early 1930's investigations on the subject were carried out in the United States of America, and in this country the use of S.A.E.20-grade motor oils made a significant contribution to the fuel economy figures obtained with the Vauxhall and Bedford engines. Recently there has been renewed interest in this country in low viscosity oils such as S.A.E.10 and S.A.E.20. The incentives for this are rising fuel costs and the development of more competitive conditions.

There are other fundamental reasons for the interest. One is that recent advances in lubrication technology have made it practicable to use lighter oils. Perhaps the most relevant advances are those concerned with the use of additives, because some of the problems associated with the use of lighter oils can be resolved satisfactorily only through the incorporation of additives.

A second reason for the increased interest is that the oils now available are so much more efficient than were pre-war lubricants. So far as lubricating properties are concerned, present day low viscosity oils are equivalent to the higher viscosity oils of the 1930's. Other factors that are tending to make the use of low viscosity oils more practicable are: the use of surface treatments on engine parts, such as phosphate coating and tin plating, to improve running-in properties; the adoption of design features such as taper faced piston rings, chromium plated rings, and special piston skirt contours; and the great improvements in oil and air filtration.

Desirable properties

A specification quoting only the viscosity number does not adequately indicate whether or not an oil is suitable for a specific application. Other qualities of the lubricant such as oiliness, maintenance of film strength and oxidation stability are also important. In addition, consideration must also be given to such performance features as cleanliness, oil consumption, improved wear and reduced corrosion. These may be obtained by appropriate blending and refining processes as well as by the use of suitable additives.

Experience shows that in suitable applications, appreciable economies can be obtained from the use of low viscosity oils. Fuel savings of more than 6 per cent with S.A.E. 5W and over 5 per cent with S.A.E. 10W have been quoted for bench tests. Indeed, for some tests it is claimed that the savings range from 7 to 10 per cent. It is doubtful, however, whether such savings would be obtainable in ordinary running conditions.

In a paper read before the South Wales and Monmouthshire section of the Institute of Transport, Mr. F. Lawrence of Shell-Mex and B.P. said that the gains experienced at normal temperatures were approximately equal for changes from S.A.E. 30W to S.A.E. 20/20W and from S.A.E. 20/20W to S.A.E. 10W, but that only slight further gain, or even none at all, was obtained by a change to S.A.E. 5W. It seems possible that the slight reduction in friction losses with S.A.E. 5W oil is largely offset by a reduced efficiency of the gas seal in the ring belt.

Worn engines

There are, of course, disadvantages associated with the use of thin oils. For example, there are the possibilities of increased oil consumption and increased soot contamination and oil detonation due to blow-by. These effects will be particularly noticeable in worn engines. Indeed it is doubtful whether low viscosity oils should be used in engines that have already given long service. However, some investigators report little or no increase in oil consumption or deterioration, and there is no doubt that piston ring design and arrangement have a marked effect on these aspects of the problem. Opinions differ on the subject of wear. Some investigators report that the additional wear is negligible; others report that although the wear is more severe, this is more than offset by fuel and other economies.

Operating conditions

Operating conditions are important. For example, commercial vehicles on short runs seldom reach maximum operating oil temperatures and for these it would probably be safe to use low viscosity lubricants. There would almost certainly be fuel economy, particularly where many starts are made from cold. Even greater benefits could probably be obtained on long distance services, if there were some provision to prevent the oil temperature from exceeding a specified maximum. Other operating conditions that affect the choice of a lubricant are the nature of the terrain, the degree of wear in the engine, the amount of oil dilution with fuel normally experienced, and the degree in which carbon contamination tends to thicken the oil. Ambient air temperature is also important since it has a marked effect on sump temperature. A great deal of guesswork could be eliminated if records were made of oil temperatures

under all running conditions during different seasons and in different countries.

Design factors are also important. Oil temperature, for example, is governed by, among other things, sump design, the use of oil coolers, and the position of the sump in relation to the radiator and the exhaust pipe. It would appear that when low viscosity oils are employed, babbit-lined bearing shells are better than lead bronze shells. Bearing clearances also need careful investigation, since oil starvation in remote parts of the lubrication system could occur as a result of excessive leakage through the bearings nearest to the source of supply. The oil pump capacity must also be considered. In general, however, the oil pumps fitted to modern engines have delivery capacities adequate to cater for the lower viscosity oils.

With the lighter lubricants, filtration of both oil and air is most important, since the film thickness between bearing surfaces is less than with the heavier oils. Therefore smaller particles of foreign matter will bridge the oil-filled gap between the surfaces, to cause scoring and abrasion.

As friction increases with the relative velocity of the moving parts, it follows that greater benefits from the use of thin oils will be obtained with high speed engines rather than with the slower engines. Moreover, the higher frequency and shorter duration of the power impulses in high speed engines, as well as the stronger hydrodynamic lubrication effect, tends to make them the more suitable for operation with low viscosity oils.

Cylinder wear

It is generally agreed that most cylinder wear takes place during starting from cold, and there is little doubt that the thinner oils will more quickly wet the cylinder walls. This will in some degree offset any possibility of wear through film breakdown. It is open to debate whether the oil of low S.A.E. numbers will run off the cylinder walls more quickly and so nullify the advantages obtained in starting from cold. This can be proved only by further experience. Easier starting should lead to increased battery and starter life.

Most of the experimental work so far carried out has been concerned with the use of lighter oils in existing designs. Much more satisfactory results should be obtained with engines specifically designed for low viscosity oils. For example, some modification of piston ring arrangement will probably be necessary, since there is only a small margin between having adequate lubrication with a low rate of wear and undue oil consumption owing to leakage past the rings.

Design factors

Sump design must also be studied. Unduly high oil temperatures must be avoided, and corrugation of finning might be necessary. A blister-type cast sump with a baffle appropriately placed above the oil level to separate it from the crankcase, would closely resemble a dry sump system. More positive means of directing cool air over the sump and of preventing warm air that has passed through the radiator from passing over it would be advantageous. A simple baffle would probably be adequate for this purpose.

Even the use of an oil cooler might be considered. Possibly the least expensive way of introducing such a unit would be to employ a by-pass system, on the lines already used for oil filters, and to use a water jacketed cooler cast integral with the cylinder block. This would avoid the expense of a separate pump and filter unit, and advantage could be taken of the less extreme range of temperatures obtained with a water cooled system. Of course, the coolant flow would have to be such that the in-going cool water passed first through the oil cooler jacket.

Bearing clearances would probably be reduced, since excessive flow through some bearings could lead to starvation of others. Furthermore, small clearances are probably desirable for maximum bearing efficiency. With smaller clearances, rigidity, especially of the crankcase, would have added importance. Attention to all these factors is essential if the optimum results are to be obtained from the use of low viscosity oils.

There is an unfortunate tendency in some quarters to expect more from a development than the facts warrant. This may well happen when low viscosity oils are used for the first time. It must therefore, be stressed that even under the most favourable conditions, the substitution of light oils for those now in general use can scarcely be expected to lead to marked improvements in efficiency with a consequent marked decrease in fuel consumption.

Nevertheless, all the evidence suggests that definite, though slight benefits may be expected. Present-day power units and transmissions have reached such a degree of efficiency that there is little hope that any one factor will greatly reduce operating costs. Any marked improvement can come only as a result of the cumulative effect of many slight improvements.

CONTENTS

SEDDON CHASSIS

A Range of Diesel-engined Goods and Passenger Vehicles

WHAT may best be described as traditional British heavy vehicle design applied to a series of medium-capacity chassis is the distinguishing characteristic of Seddon Diesel goods and passenger machines. From the outset these vehicles have employed Perkins engines as power units. In this connection it must be recalled that the modern type of British automotive diesel engine came into being at a time of severe economic stress, and that, to a very great extent, its rapid development was a result of the pressure of a period when all the emphasis was placed upon reduction of operating costs.

Because of that state of affairs, the oil engine was at first chiefly applied to the largest class of vehicle, since its economy was only fully exploited under maximum load conditions in continuous operation. However, after the initial development troubles had been overcome it became apparent that fuel economy was not the only virtue of the type. Dimensional and material specifications had necessarily been adopted that provided both mechanical reliability and longevity. Moreover, when the combustion process and its associated problems were properly understood it was appreciated that the thermal cycle itself had an inherent influence in the same direction.

Smaller diesel engines, such as would be suitable for medium - capacity vehicles, were not at once developed with the same enthusiasm and enterprise. The then current opinion was that as medium-capacity vehicles of about five tons payload were derived mainly from the private car side of the industry, their chassis were of relatively light construction, particularly at the front end, and that in consequence their life was comparatively short — much shorter, indeed, than the life of a diesel engine. This state of affairs created the uneconomic situation of reversing the customary service procedure of fitting replacement engines in part-worn chassis to the need for building replacement chassis around a less than half-worn engine.

Nevertheless, where such medium-capacity trucks could be operated intensively on long-distance haulage, coupled with a greater or less degree of overload, the saving in fuel cost could more than off-set the comparatively low cost of a new chassis, less engine. This procedure, in which an engine was transferred through a sequence of chassis, was profitable to some operators, even if it was uneconomic in a wider sense. It led to a brisk business in the conversion of existing medium-capacity trucks to diesel power as soon as a proprietary engine of suitable size became available.

RANGE OF CHASSIS

Model	Wheelbase	Weight	Capacity
5L Goods	13 ft 6 in	48 cwt	6-7 ton
5S 10 Tipper	10 ft	50 cwt	6-7 ton
5S9 2 Tractor	9 ft	47 cwt	10 ton
8 Tractor*	9 ft	48 cwt	10 ton
4C Goods	14 ft 11 in	58 cwt	6-7 ton
4 Passenger	14 ft 11 in	58 cwt	32 seat
6 Passenger	16 ft 4 in	60 cwt	35 seat
7L Goods	10 ft 6 in	36 cwt	3 ton
7S8 Tipper	8 ft 6 in	39 cwt	3 ton
7S6 Tractor	6 ft 10 in	30 cwt	8 ton
7P Passenger	10 ft 6 in	37 cwt	25 seat

**Equipped for Scammell semi-trailers*

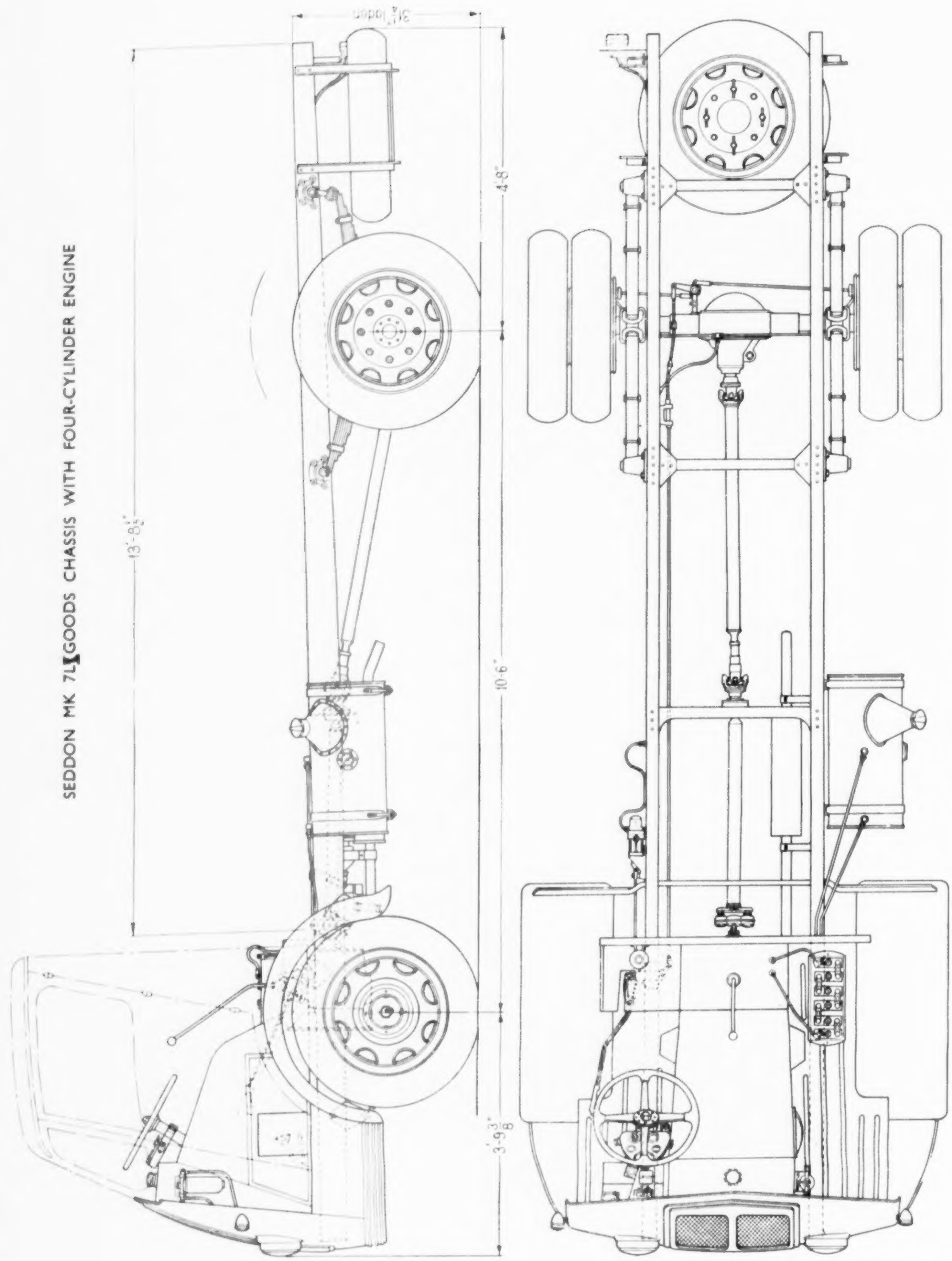
The obvious need, however, was for a medium-capacity chassis built upon the same lines as the heavy transport vehicles and specifically laid out to take full advantage of the economic influences upon design arising from the characteristics inherent to the diesel engine. This was the idea behind the origin and production of the Seddon range.

Perkins engines, being the only proprietary make of the required size, were naturally adopted and have been used exclusively ever since. Similarly, the other main components were proprietary units, selected for their robustness of design, and embodied in a general layout that was simple and straightforward. Appropriate constructional methods were followed, notably in the use of fitted bolts in the assembly of the main structure, and in adherence to a rectangular frame plan, incorporating side members as free from upswept arches and inward crankings as possible. Thus, throughout the entire series of Seddon vehicles there is a recognizable unity of design and an absence of non-essentials characteristic of the typical British transport vehicle.

At the present time the Seddon range comprises eleven chassis, eight for goods and three for passenger work. The basic model is a normal lorry chassis from which are derived variants as to wheelbase, overall dimensions and load capacity, providing for 3, 5 or 7 tons payload. There are also two short wheelbase models for end-tipping applications and three tractors, one for large articulated six-wheelers of permanently-coupled type, another with its chassis modified to incorporate Scammell automatic coupling and a third which is a small unit suitable for the local transport class of semi-trailer.

Mk 5L goods chassis with two-speed axle. The general layout is basically common to the whole range

On the passenger chassis, two are for 32 or 35 seat capacity single-deck bodies, their respective wheelbase lengths being 14 ft 11 in and 16 ft 4 in. The third is a short-wheelbase (10 ft 6 in) model



SEDDON MK 7L GOODS CHASSIS WITH FOUR-CYLINDER ENGINE

for a 25-seater one-man operated bus. This is the smallest oil-engined public passenger transport vehicle built in this or any other country. All chassis, both goods and passenger, are powered by Perkins P6 engines except the 3-ton goods and the 25-seater passenger machines, which have the P4 unit.

The frames on which the goods chassis are based are of rectangular form, of rolled channel steel, with level top flange. The maximum depth of section is $8\frac{1}{8}$ in, tapering to the front to $5\frac{1}{8}$ in and to $4\frac{1}{8}$ in to the rear. Passenger frames differ from the others in that they incorporate shallow swept wheel arches and are cranked inwards to reduce the overall width of the load-carrying portion from 44 in to 33 in over the front axle. The greatest depth of the channel section is $8\frac{1}{8}$ in and this tapers in front to $5\frac{13}{64}$ in and to $4\frac{1}{8}$ in at the rear. The cranked part of the side member is reinforced by a plate welded across the flanges to form a box section, the ends of this piece being notched to graduate the change. Tapering of the side members is effected by bringing up the lower flange while the upper one is level throughout its entire length except at the wheel arch. The flanges are $2\frac{1}{4}$ in wide. Alloy steel (V6A) of $\frac{1}{4}$ in thickness and having a 35 ton tensile rating and 23 ton yield point is used for the side members.

The 3-ton goods and 25-seater passenger chassis also have frame side members of V6A. They are of somewhat shallower section, with narrower flanges and reduced thickness of material. The bus chassis frame is 6 in deep, tapering to 5 in in front and 4 in at the rear. Flanges are $2\frac{1}{2}$ in wide and the thickness of the metal is $\frac{3}{16}$ in.

Cross-members vary somewhat according to chassis type and also with their individual location and function; nevertheless a common plan is followed throughout. On all models there is a straight channel across the front which carries the front bracket of the three-point engine mounting and also supports the radiator. Next, and in line with the flywheel housing, is a dropped member fabricated from two steel channels set web to web but separated by welded tubular distance pieces. The resultant modified H-section, bent into a wide U form, has fabricated "top hat" section brackets welded into its ends and bolted inside the frame channels. This member contributes greatly to the rigidity of the frame and completely relieves the engine itself of any duty as a frame stiffener.

On the longer chassis a deep pressed-steel cross-member is located near the centre of the frames for carrying the intermediate bearing of the divided propeller shaft. For lightness the web of this member is extensively cut away. On the shorter chassis in which a single undivided shaft is used, this type of cross-member is not necessary.

On the goods chassis in general, the cross-members are of channel section of similar material to the frame. They are located between the rear spring brackets, fore and aft of the axle, and are substantially gusseted and riveted to the top and bottom flanges of the side members. The equivalent cross-members on the larger passenger chassis are tubular, being $3\frac{7}{8}$ in and 5 in in diameter with $\frac{1}{4}$ in wall thickness. They have machined and welded flanges bolted through the side member web, the bolts picking up the spring brackets mounted on the outer face of the channels. There is also a 3 in 12 gauge cross tube at the extreme rear of the frame, while on the longest bus chassis (Mk. 6) an additional arched cross tube is placed behind the engine bearer cross-member and forward of the intermediate bearing support.

With the exception of the two

SPECIFICATION

ENGINE: Six- (or four-) cylinder oil engine with air-cell combustion chamber. Bore and stroke, 88·9 mm × 127 mm, 4,730 c.c. (or 3,150 c.c.). Seven (or five) bearing crankshaft with steel shell lead-bronze bearings. C.A.V. injection pump with two-hole sprayers and pneumatic governor.

CLUTCH: Single dry plate, 12 in (or 10 in) diameter.

GEAR BOX: Five-speed with single helical constant mesh gears on top, fourth and third ratios. Direct top or overdrive (optional). Models with four-cylinder engine have four-speed box with helical constant mesh top, third and second speed.

TRANSMISSION: Open tubular propeller shaft, divided or single according to wheelbase. Divided shafts have rubber bushed flexible joint at gear box coupling otherwise needle-roller universals throughout.

REAR AXLE: Hypoid bevel, four-pinion differential. Full floating. Taper roller bearing hubs. Ratio, 6·16 to 1 (or alternatives). Two-speed axle 6·33 and 8·81 to 1) on tractor models (also applicable to other heavy duty types as optional extra).

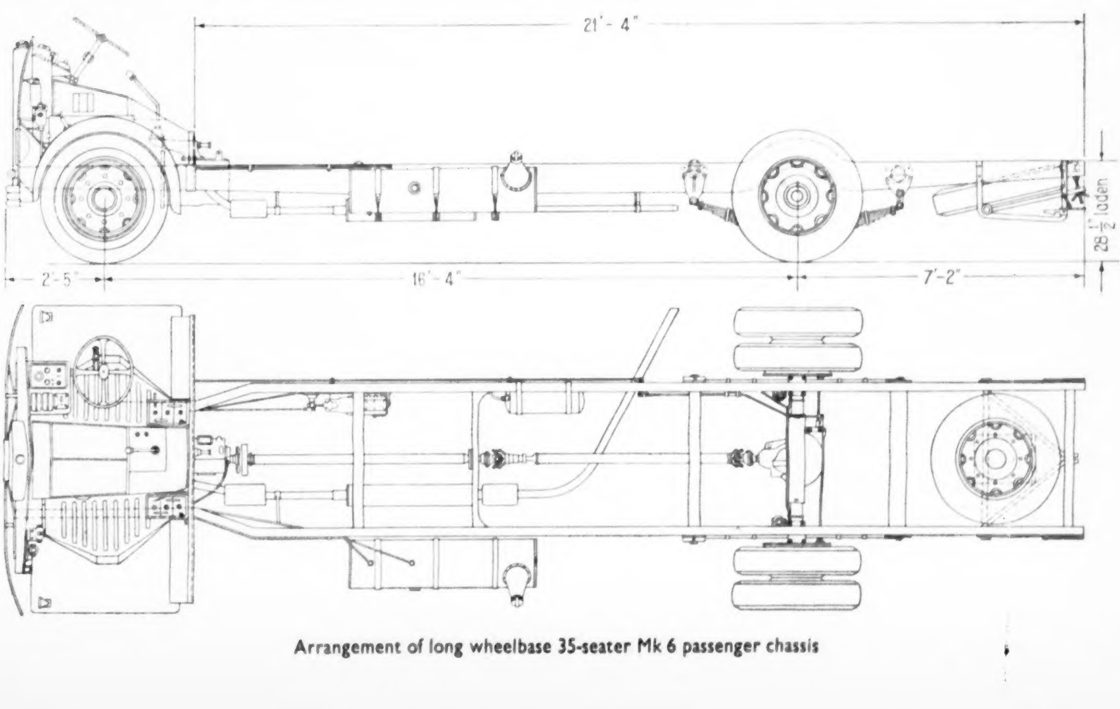
STEERING: Cam and twin roller.

SUSPENSION: Semi-elliptic springs. Helper leaves on goods models. Hydraulic shock dampers on passenger chassis.

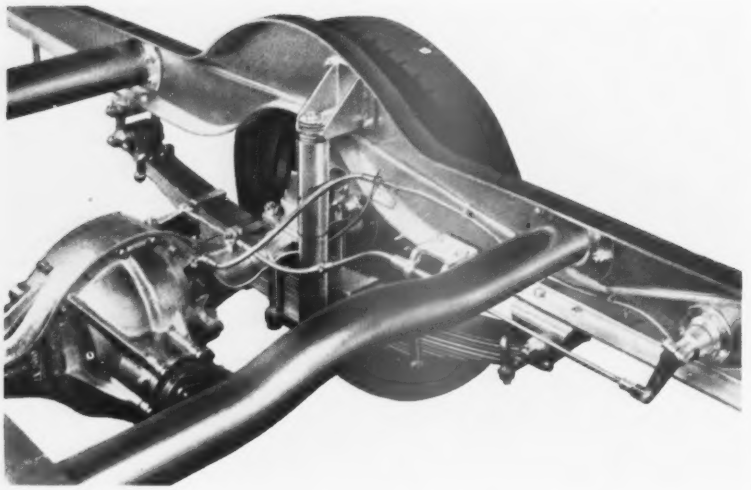
FRAME: Pressed steel channels and cross-members, also tubular cross-members on passenger models.

BRAKES: Hydraulically operated two-LS type. Vacuum servo assistance on six-cylinder engined models.

ELECTRICAL EQUIPMENT: C.A.V. 12 v lighting and starting.



Arrangement of long wheelbase 35-seater Mk 6 passenger chassis



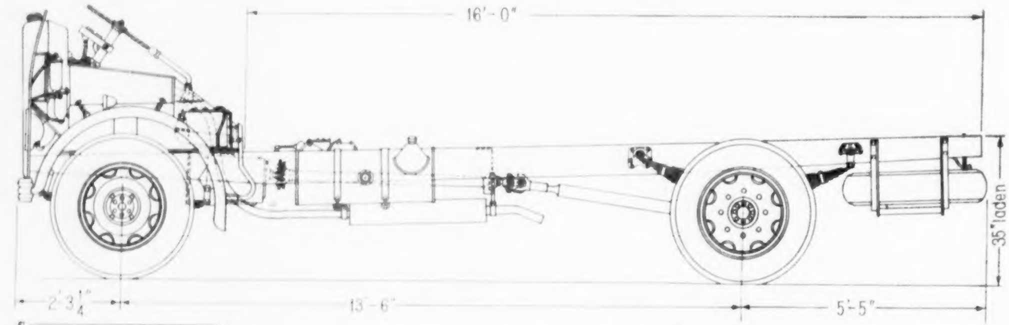
Tubular cross members and swept wheel arch on Mk 4 and Mk 6 passenger chassis

models powered by the four-cylinder engine, all chassis follow the conventional forward-control layout with driver beside engine, the power unit being mounted directly above the front axle. In the case of the two exceptions the forward overhang is greatly increased so that the engine and gear box are actually forward of the axle, thus permitting the driver's cab on the goods chassis, or the passenger entrance on the bus to be forward of the front wheels.

An interesting feature of frame construction is that the side members are jig drilled and are thus interchangeable and replaceable without adjustment of bolt holes or the like "fitting" to suit the existing components. It may be considered that such a state of affairs could be taken for granted, but it does not follow that it universally obtains in heavy vehicle manufacture. However, although Seddon output has never claimed mass-production status, the fundamental requirements have always been fully recognized, hence one of the earliest pieces of special equipment laid down was a side member drilling rig consisting of a jig fixture incorporating a railed track along one side of its length. A small trolley carrying a permanently mounted electric drill traverses this track, adjustment for drill spindle height being provided so that the drill can be aligned with the bushed jig plates attached to the fixture on which the channel is mounted. By limitation of bolt sizes and by the disposition of their location in the web, frame drilling on this simple shop-made rig is carried out rapidly and with as high a degree of accuracy as on some of the exceedingly elaborate machine tool equipment now available.

Semi-elliptic springs of 90 ton oil-hardened silicon - manganese steel (EN45) are used for both front and rear suspension. There are variations in mounting and dimensions in the different models. For example, the loaded between-centres length is from 38 in to 45 in (front), and from 52 in to 60 in (rear). A small camber is maintained on all front springs, but at the rear this varies with the type of vehicle, the tendency being to approach flatness under load, or even to pass into slight reverse camber. This latter condition is most evident on the larger passenger models, on which, incidentally, the front deflection is 2$\frac{5}{8}$ in and the rear 5 in. On these chassis front and rear suspension is controlled by Newton telescopic dampers while on the small bus, Girling horizontal link-connected double-piston dampers are fitted. All passenger models are provided with rubber bump stops for both front and rear axles.

Spring leaves vary in number, width and thickness according to the vehicle type, its load capacity and the conditions prevailing in the country in which it will be used. On the heavier goods and the larger passenger chassis the spring leaves are 3$\frac{1}{2}$ in wide on both axles while on the 3-ton and 25-seater types they are 2$\frac{1}{2}$ in. A typical spring make-up of the heavy type is 11 leaves for the front and 15 for the rear, with a $\frac{1}{8}$ in thick main plate. The second and other supporting leaves may also be of the same thickness, the



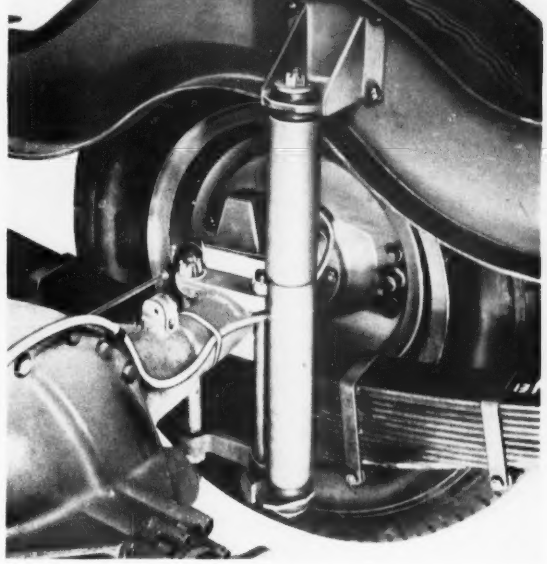
Mk 5L goods chassis, the basic Seddon type

remainder of the assembly being made up of 5/16 in leaves. As there are some 70 suspension variations it is not possible to particularize more precisely.

There is, likewise, a varied employment of rebound plates, anti-roll clips and extended second leaves to reinforce the main leaf eyes. In general, the heavier goods models incorporate full-length rebound plates and also have "helper" leaves which give progressive suspension characteristics under maximum load conditions. Passenger models have short anti-roll plates instead of the full-length type.

Shackle pins are of the headless parallel type, of case-hardened alloy steel (EN33), clamped in the split ends of the shackles, the pinch bolts engaging cotter grooves in the pins for lateral security. The same method of fixing in the outer faces of the frame brackets is used, the inner ends of the pins being supported in plain push-fit holes. Shackle-pin diameter is ¾ in (front) and 1 in (rear) on all models except the two large passenger chassis and the extra long wheelbase (4C) goods model based on them, which have 1¼ in pins all round.

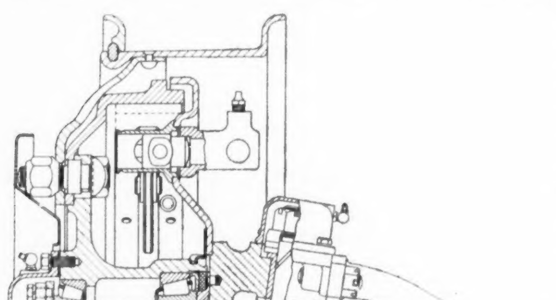
Attachment of the springs to the axles also shows some variation of method from one type of chassis to another. All front springs are above the axle beam, with U-bolt fixing, while the rear springs are above the axles on the goods chassis but are underslung on the two larger passenger models, the Mk. 4 and Mk. 6 on which the springs are directly under the frame side members, whereas on the goods types they are outside the frame. Springs outside the frame and resting on the axle spring seatings are held by inclined U-bolts but the underslung springs of the passenger models are held by four hex - head high-tensile bolts and a clamping plate.



Underslung rear spring and Newton shock damper on Mk 4 and Mk 6 passenger chassis

Front axles are of standard proprietary type, the forged beam being of oil - hardened chrome - molybdenum steel of 48 tons tensile. Stub axle forgings are of similar material with the tensile raised into the 50-60 ton range. The forged steel hubs run on Timken taper roller bearings. Rear axles also show some divergences, not only from the final drive aspect, which will be dealt with more fully later, but also structurally. In short there is no commitment either to the integral one-piece axle beam or to the composite type with tubes inserted into a cast differential casing, both constructions being used.

The power unit common to the greater part of the range of chassis is the Perkins P6 oil engine, the exceptions being the 3-ton goods and 25-seater passenger machines, which are powered by the P4. Both engines are of the same general type and apart from the number of cylinders, are of similar specification, of monobloc construction and having the same cylinder dimensions of 3½ in bore and 5 in stroke (88·9 mm × 127 mm, 4,730 c.c.). Normally the P6 engine is rated at 70 b.h.p. at 2,200 r.p.m., with maximum torque of 184 lb-ft at 1,000 r.p.m. but as installed in Seddon chassis the output is increased to 79 b.h.p. by raising the governed speed to 2,400 r.p.m. Similarly the 46 b.h.p. (2,200 r.p.m.) normal output of the 3·15 litre P4 engine is stepped up to 50 b.h.p.



Front axle steering swivel and hub assembly. Common to all models with six-cylinder engines

The combustion system employed in these engines is the Perkins "Aeroflow" type with off-set air-cell combustion chamber and a two-hole C.A.V. injector, one spray from which is directed into the cell while the other is aimed into the throat of the venturi connecting the cell with the clearance space. Compression ratio is 16·5 to 1 and injection pressure is 1,800 lb/sq in, the pump being the C.A.V. type with integral pneumatic governor. The forged chrome-molybdenum crankshaft is supported by steel shell lead-bronze bearings between each throw, end thrust being taken in both directions by the side faces of the rear bearing. The journal diameter is 70 mm, and the crankpin diameter 57 mm.

Dry liners are fitted in the cast iron block. The head is also of cast iron and in one piece, covering six (or four) cylinders. The vertical overhead valves are push-rod operated through rockers from a high camshaft running in plain bearings in a tunnel on the right side of the cylinder block immediately beneath the head joint. The camshaft is driven by a triple roller chain which also drives the injection pump. There is a single V-belt for the water impeller and fan, which are mounted on a common spindle running in a ball and roller bearing mounted side by side. Belt tension is adjusted by movement of the dynamo, its pulley serving as a jockey.

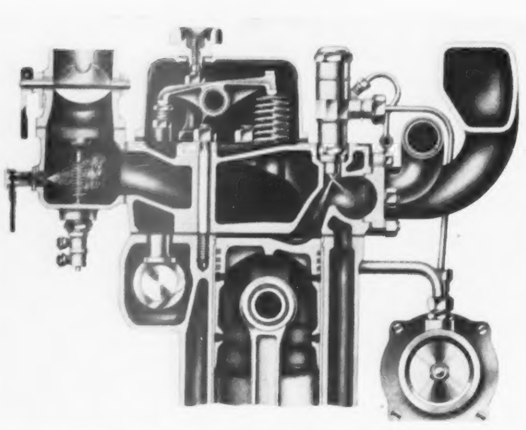
The electrical system operates at 12 volts, as distinct from the 24 v generally adopted for the British diesel. It may be pointed out, however, that under normal conditions in this country, even in winter, the 12 v equipment is adequate for starting and its use makes an important contribution to weight reduction. For more difficult

conditions, Perkins engines incorporate, as standard fitments, the Kigass system of paraffin injection into the inlet manifold, in which there is also an electrical heater coil.

Pressure lubrication is provided by a slow-running rotary oil pump driven from the shaft of the injection pump coupling through steel skew gears. The low pump speed ensures efficient self-priming. Oil is drawn from the sump through a large strainer and is delivered by way of a Vokes full-flow filter, flange-mounted on the left side of the crankcase. Coolant circulation in the cylinder head is assisted by the impeller which rotates in a flange-fitted housing on the front face. No positive flow is directed through the cylinder jackets, cooling round the bores being by natural circulation.

On the P6 engine the injection pump is driven in tandem through a Clayton Dewandre vane-type vacuum brake exhauster. The shortness of the four-cylinder (P4) unit precludes this arrangement and in consequence the vehicles fitted with this engine are not equipped with vacuum servo brake actuation.

Engine mounting on all models provides limited flexibility through the medium of bonded rubber bushings and pads with three-point support. At the rear two mountings are attached to the flywheel housing to coincide with the end brackets of the fabricated dropped cross-member already described. In front a bracket is bolted across the cylinder block to co-operate with a rubber support of the sandwich type resting on the front cross-member. The detachable tube radiator also stands on rubber pads on the same member. It is positioned by stays at the top on all models except the Mk. 7 types on which the radiator is stayed in the upright position by bonded rubber sandwich brackets connected between its side frames and the fabricated steel apron plate mounted across the front of the chassis.

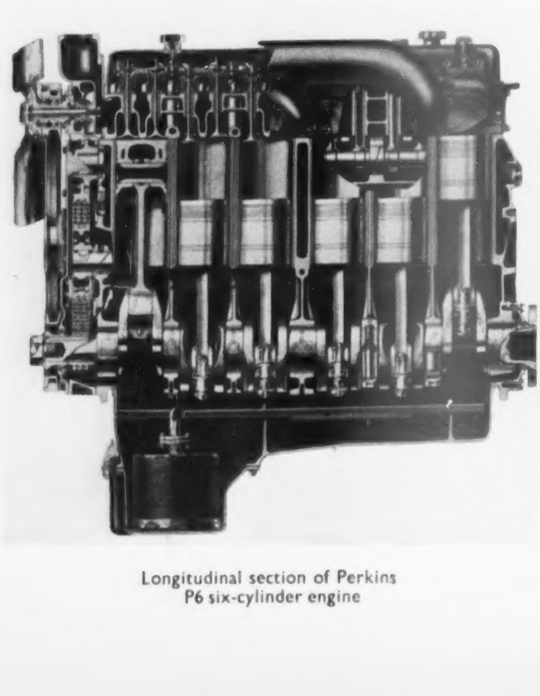


Perkins cylinder head, showing Aeroflow combustion chamber and Kigass starting equipment

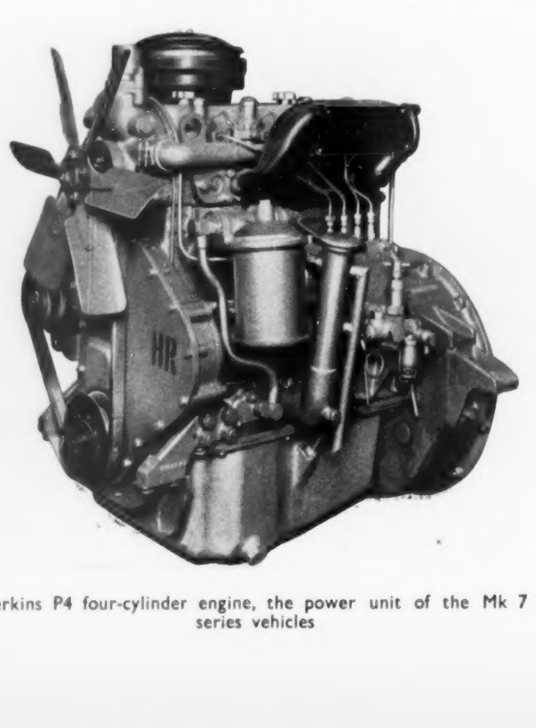
The gear box is coupled to the engine through a cast aluminium bell-housing which encloses the Borg & Beck clutch. On the larger engine this is 12 in diameter with 151 sq in frictional area. Its torque capacity is 203 lb-ft at 1,500 r.p.m. The Ferodo friction material is ¼ in thick. Withdrawal is through a deep-groove ball bearing. On the P4 engine the clutch diameter is 10 in, and the total frictional area is 82 sq in. This clutch will transmit a torque of 123 lb-ft at 1,500 r.p.m.

Proprietary gear boxes are employed. Normal equipment for the larger models is a five-speed type with direct drive on top gear; there is an alternative overdrive top for special requirements. These boxes were designed in the first instance specifically to Seddon requirements. Four-speed boxes are also available and are the standard equipment on the small chassis with four-cylinder engines. Gears are of nickel-chrome case-hardened steel. The main shaft is short and stiff, the root diameter of the splined portion being 1⅝ in on the five-speed type. On this box the construction is somewhat unusual in that the cast aluminium casing is in two halves, the joint being vertical through the centre line of the two shafts which are carried in ball bearings throughout. The outer races of the bearings are seated in machined steel cups secured in position by stud-fixed cast caps, being registered in holes machined in the ends of the assembled gear box casing but not specifically clamped therein by the drawing together of the two halves by the transverse bolts which pass through the assembly.

Both shafts are supported by intermediate bearings mounted in a web or partition which divides the interior of the box into two parts, one housing the layshaft constant mesh pinions and fourth gear and the other the three lower ratios. The adequate support provided for the main shaft eliminates the necessity for location by a spigot bearing in the first-motion shaft. This latter is located by the combination of a ball and a roller bearing in which it



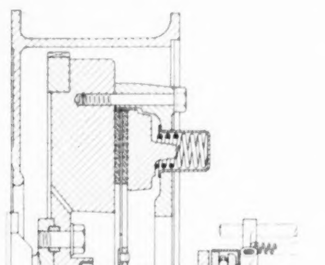
Longitudinal section of Perkins P6 six-cylinder engine



Perkins P4 four-cylinder engine, the power unit of the Mk 7 series vehicles

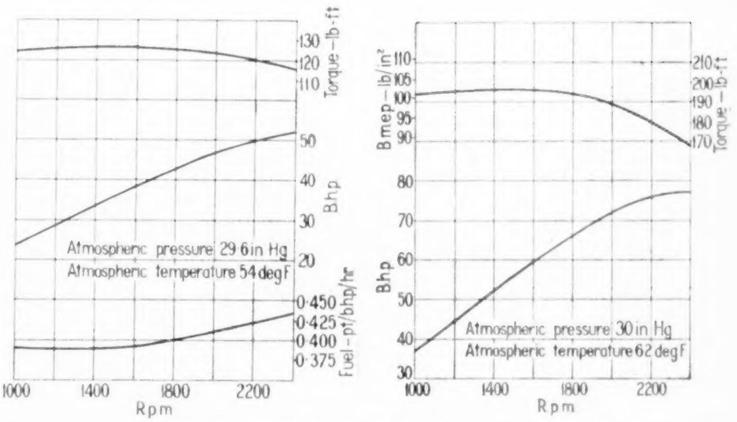
is carried. Gear engagement is by sliding dogs for the three higher ratios which have constant mesh helical pinions. The two lower ratios have sliding engagement of their straight-tooth gears. Only the first-and-reverse gear pinion is integral with the lay-shaft. All the other pinions are pressed on separately. They are located and driven through long keys. Control is from a "remote" ball-change mounted directly on the gear box. An extension housing for the gear lever pivot projects forward and the lever itself is also bent forward, so that the knob is placed above the flywheel housing.

An interesting detail of assembly practice, which again reflects the Seddon approach to these matters, is seen in a rig for bringing the engine and gear box together. It comprises a fixed stand, extending from which is a short track on which a trolley moves to or from the flywheel end of the mounted engine. A fixture on the trolley that can be raised or lowered by a small hydraulic jack receives the gear box in a position in which its input shaft is in the same vertical plane as the engine crankshaft. Lateral play in the trolley runners and height adjustment by the jack enable the gear box to be offered up to the flywheel housing with the clutch shaft spigot accurately positioned, so avoiding any risk of distortion or stress on the spigot bearing such as may occur in carrying out this operation by block and tackle methods.



Borg and Beck 12 in clutch in P6 engine flywheel

Normal gear ratios are 1, 1·6, 2·79, 4·73 and 8·14 to 1 forward, with 7·76 to 1 reverse. In conjunction with the standardized axle ratio of 6·16 to 1, a road speed of 42·20 m.p.h. is attained at the governed engine speed of 2,400 r.p.m. For certain classes of operation, such as trunk route long-distance haulage, or touring coach work, a higher speed and improved fuel consumption may be gained by use of a higher top gear ratio and in such circumstances the indirect overdrive provides ratios of 0·812, 1, 1·72, 3·65 and 6·61 to 1 forward, with 6·38 to 1 reverse. With this box and the same axle ratio and engine speed, the road speed is 52·71 m.p.h.



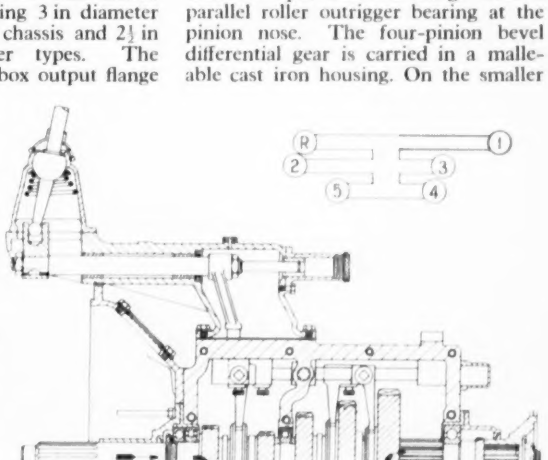
Performance curves of Perkins P4 (left) and P6 (right) engines fitted to Seddon vehicles. Normal output is increased by raising maximum r.p.m. from 2,200 to 2,400

The four-speed box fitted to the smaller chassis has ratios of 1, 1·78, 3·17 and 5·72 to 1 forward and 6·86 to 1 reverse. At the same governed engine speed of 2,400 r.p.m. and with an axle ratio of 6·83 to 1, the maximum road speed is 38-39 m.p.h. or 44·0 m.p.h. with a 5·57 to 1 axle.

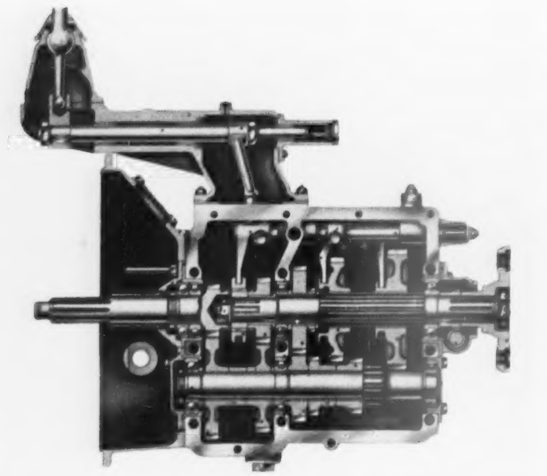
Transmission is by open tubular divided propeller shafts on all except the very short-wheelbase and tractor models, the shafts being 3 in diameter on the six-cylindered chassis and 2½ in on the four-cylinder types. The coupling at the gear box output flange is the Layrub Type 90, on models with the P6 engine and Type 80 with the P4 unit. Support for the outer end of the primary propeller shaft is given by a self-aligning bearing assembly rigidly mounted in the web of the deep channel cross-member fitted on all the longer wheel-base models. The rearward portion of the divided propeller shaft has Hardy-Spicer Type 1510 universal joints at each end on the larger and Type 1410 on the smaller - engined chassis. Models with single propeller shafts have joints of the mechanical type at each end of the shaft.

As has already been mentioned, there is some divergence in rear axle type throughout the range. The usual equipment is now the Moss hypoid bevel pattern which has replaced the spiral bevel. On the heavier goods and tractor chassis there is also available the two-speed final drive provided by the Eaton axle, which had its first British application on Seddon vehicles.

For normal service the Moss axle has a reduction of 6·16 to 1. It is of the full-floating type with 1 [illegible] in drive shafts of oil-hardened 75 ton steel (EN25), which have their outer ends upset to form coupling flanges to which the hubs are attached by eight ½ in high tensile studs and nuts. The hypoid bevel pinion is of 3 per cent nickel-chrome steel and is straddle mounted, running in two steep-angle Timken taper roller bearings with a parallel roller outrigger bearing at the pinion nose. The four-pinion bevel differential gear is carried in a malleable cast iron housing. On the smaller



Diagrammatic arrangement of overdrive five-speed gearbox, first gear engaged



Five-speed gearbox. The casing is divided vertically on the axes of the shafts

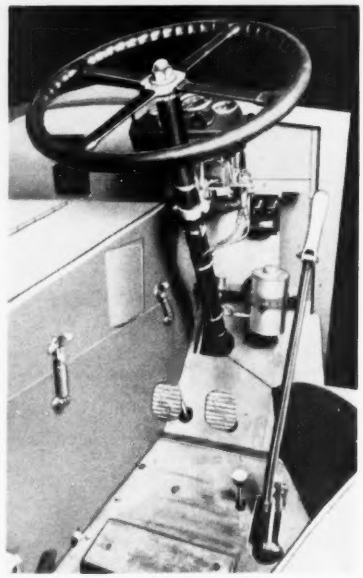
four-cylinder engined chassis the rear axle is of the same general design except that the drive shafts are reduced to $1\frac{1}{2}$ in diameter.

In the Eaton axle, in which the main reduction is by spiral bevel, the two ratios provided are 6·33 and 8·81 to 1. The lower ratio is obtained through an internally toothed epicyclic gear interposed between the crown wheel and the differential cage, and selection is effected by movement of a dog clutch to lock the rotating parts together as a solid unit or to hold the reaction member (the sun pinion) against rotation by connecting it to the axle casing. Dog clutch movement is made through a vacuum-operated diaphragm unit on the axle controlled by flexible cable from a button attached to the normal gear lever. The axle shafts in this unit are of EN25Z steel and though approximately of the same diameter they are treated to give 100 tons tensile to cope with the increased torque loading when operation is through the lower final drive ratio.

On all models the hubs run on taper roller bearings and are provided with eight $\frac{7}{8}$ in wheel studs; the brake drums are fitted on the same studs, being clamped between the wheel disc and the hub body. Concentricity is assured by the machined part of the hub centre acting as a spigot on which the drum is registered; the studs are also shouldered to coincide with the brake drum holes.

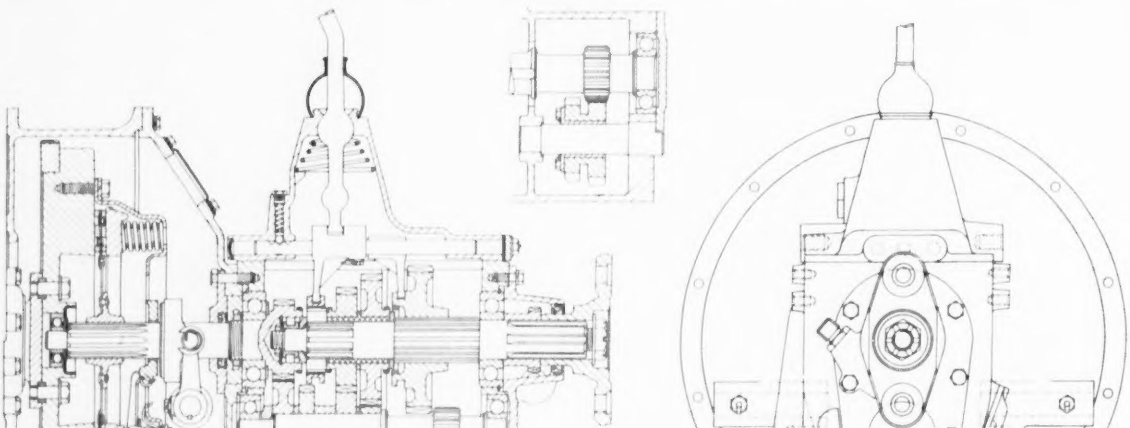
Brake drum material is alloy iron and two cooling fins and a thickened rim are embodied in the casting. On the larger chassis the internal diameter is 16 in on both front and rear axles with front shoe width of $2\frac{1}{2}$ in and rear $3\frac{1}{2}$ in. The total friction area is $376\frac{1}{2}$ sq in; lining thickness is $\frac{3}{8}$ in. On the smaller machines the rear brakes are also 16 in diameter, but the front are 14 in, all shoes are $2\frac{1}{2}$ in wide and total braking area is 292 sq in.

Brake shoe mechanism throughout the entire range is of the Girling two-leading-shoe type, hydraulically operated and with vacuum servo assistance except on those chassis with the P4 engine. On the larger models the brake actuating mechanism is combined in a single unit comprising the Clayton Dewandre vacuum servo cylinder and the Lockheed (or Girling) hydraulic master cylinder. On passenger chassis this assembly is mounted on a steel plate across but not in contact with the flanges of the channel inside the frame. It is held by two bolts which pass through the side member web on its longitudinal centre line, suitable distance pieces holding it just clear of the flanges in order to avoid the effect of a local change of section from



Driving controls, instruments and switchgear on the larger vehicles

channel to box at a point where the risk of stress concentration is considerable. Also carried on the same plate is the bearing boss for the rocking lever through which the control



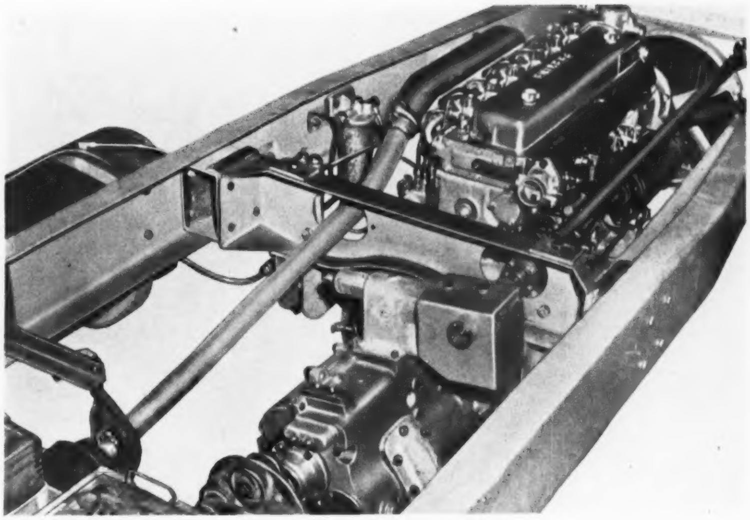
Four-speed gearbox with 10 in clutch used on Mk 7 series with four-cylinder engines

rod from the brake pedal is connected to the servo mechanism. On the goods models the brake servo unit is outside the frame (which is narrower) and is bolted directly to the web of the channel.

A 1,000 cu in vacuum reservoir is fitted in a location to suit the layout of the particular chassis type (the same consideration also applies to the cylindrical 25 gallon fuel tank). Mechanical actuation of the rear brakes is by rod to a relay lever adjacent to the forward bracket of the rear springs, the final connection being by flexible cable to the usual Girling transverse compensating linkage. Here again the hand brake rod may be inside or outside the chassis frame according to the layout of the particular model concerned. On the four-cylinder engined chassis, brake arrangement is much the same, apart from the absence of the vacuum cylinder. The Girling hydraulic cylinder may be either outside or inside the frame, according to whether the chassis is of the right- or left-hand drive type.

Steering is by Marles cam and double roller gear mounted at the extreme front of the frame and having a trailing drag link. Thompson self-adjusting ball joints are used throughout the steering connections. The overall ratio of the gear is 24·7 to 1; this requires approximately 4¾ turns of the 20 in diameter spring spoked wheel for the full movement (70 degrees) from lock to lock. The steering gear has been selected in relation to a maximum front axle load of 3 ton 5 cwt. Both the drop arm and the rocker shaft are of manganese-molybdenum steel of 60-65 ton tensile.

Tyres on the larger models are 12-ply 8·25 in × 20 in single front and twin rear, and run at 80 lb/sq in pressure. On the smaller machines they are 10-ply 7·00 in × 20 in.

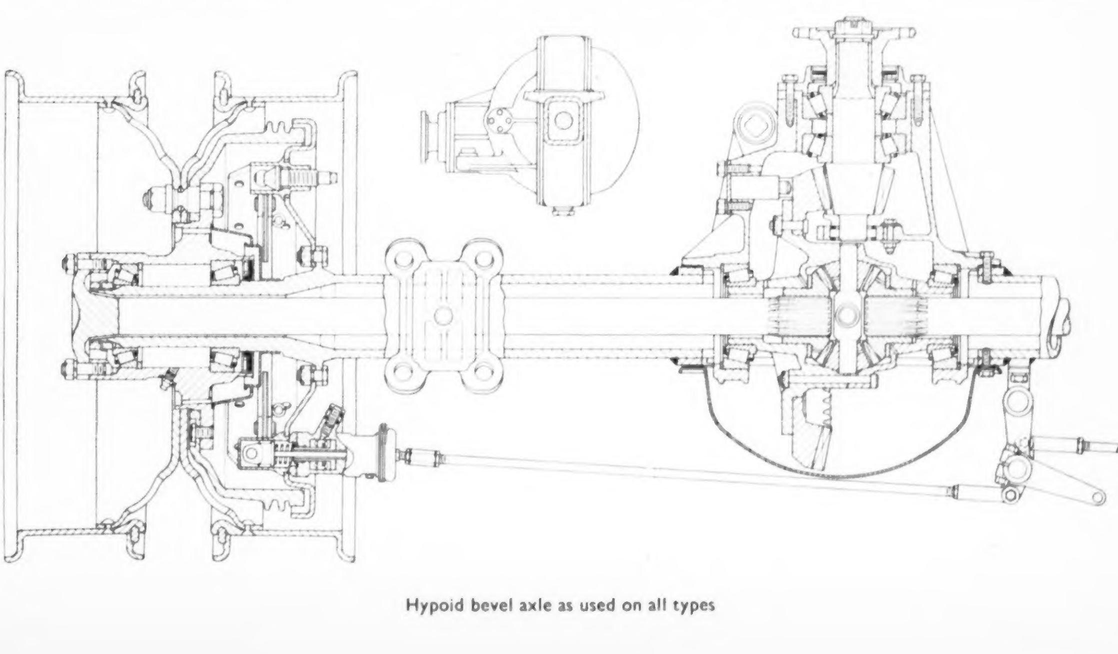


Underfloor installation of Perkins P6 engine 32-seater in single-deck chassis

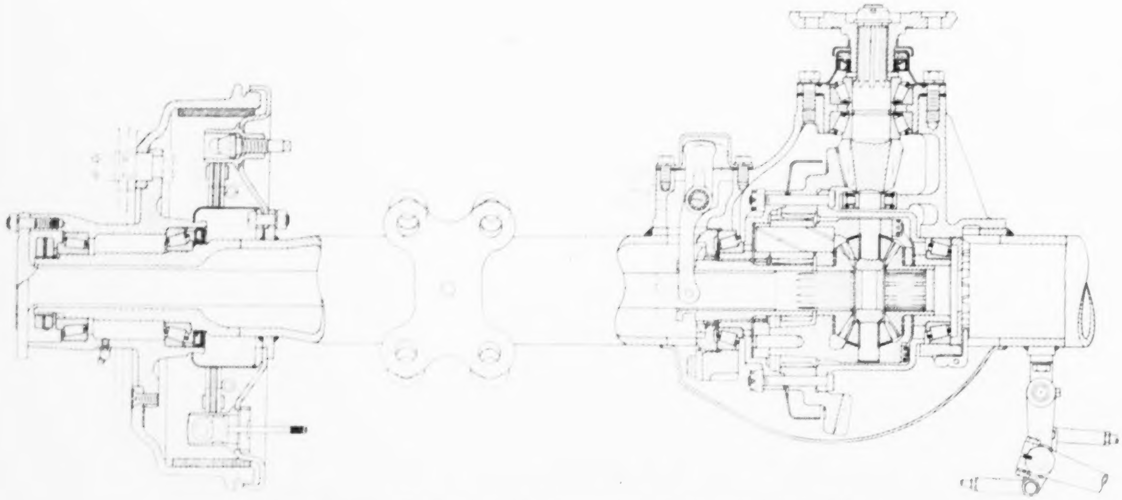
Electrical equipment on the larger models comprises a C.A.V. 12 v dynamo (D5/LF.A.10) with voltage control unit 107.A.3 and axial starter motor BS.512.K.66, in conjunction with two 17-plate 6 v 154 a.h. accumulators in series. On the smaller chassis the dynamo is the C45.PV/5 type and the batteries are of the 11-plate size, rated at 78 a.h.

All chassis are readily convertible from right- to left-hand control and can be supplied for export as bare, with cab, or with timber or metal bodies. Passenger bodies of 25, 31 or 35 seat capacity are produced in standardized form in light alloy to an exclusive design which facilitates dispatch in completely or partially knocked-down condition; in the latter state the sub-assemblies are readily brought together by semi-skilled operatives to complete a body which thereafter may be treated as a self-contained structure that can be mounted on a chassis or lifted off as a whole. The production of such a body within the same organization has resulted in the development of methods of attachment to the chassis which provide many of the advantages of integral construction, notably in the elimination of weighty and height-increasing floor bearers. A truss-plate mounted transversely across the chassis above the rear engine brackets forms a substantial reinforcement to the cab or body structure at a point where torsional deflections are liable to make themselves most evident.

A recent variation in the passenger range is an underfloor - engined



Hypoid bevel axle as used on all types



Eaton two-speed axle fitted to Mk 5 tractors and other heavy-duty vehicles

32-seater single-deck chassis which is unusual in that the Perkins P6 power unit, instead of being horizontally mounted, is of the normal vertical type. The virtually unmodified engine is positioned in the centre of the frame on dropped hangers so that its cylinder head cover does not project more than about 4 in above the level of the side member top flanges. In the complete vehicle this projection is accommodated within the body underframing by raising the floor level merely $\frac{3}{4}$ in.

The chassis follows the general specification common to the range, the only notable difference being the shortness of the wheelbase (13 ft 11 in) in relation to the overall length of 25 ft 9 in, involving a front overhang of 5 ft. Mechanically the changes resulting from the new engine position are the underslung radiator and fan unit behind the front axle and a single propeller shaft only 38 in long connecting the unit-mounted gear box to the back axle.

Access to the engine is gained through a trap in the centre gangway of the body floor, this opening being flanked by a smaller trap on the near side in which the mechanic stands with his feet on the garage floor, the chassis side member being between him and the power unit.

This unorthodox underfloor layout may be regarded as somewhat experimental in so far as the reaction of operators is concerned. The subsidiary "manhole" at the side of the engine access trap involves removal of seats, although this may be less of a problem to the small fleet owner than the provision of special pits which the flat type of underfloor installation demands for easy servicing. Complete removal of the engine necessitates the provision of an off-side central door through which the jig of a garage crane can be entered to lift and withdraw the unit. The arrangement is obviously limited to engines of moderate overall height and size, in which respect the P6 is perhaps the only suitable type of its power rating.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

The following meetings will be held during February:—

NORTH-EASTERN CENTRE

Wednesday, 18th February, 7.30 p.m. Annual General and Ordinary General Meetings in the Chemistry Lecture Theatre, The University, Leeds. Paper: "Rear Axles," by R. H. Wilson.

NORTH-WESTERN CENTRE

Wednesday, 18th February, 7.15 p.m. Annual General Meeting followed by an Ordinary General Meeting in the Engineers' Club, Albert Square, Manchester. Paper: "Safety Glass," by H. Irwin, B.Sc.(Tech.) (Man.), M.I.Mech.E.

SCOTTISH CENTRE

Monday, 16th February, 7.30 p.m. Annual General Meeting followed by an Ordinary General Meeting in the Institution of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper: "Instrumentation," by J. A. Singleton.

WESTERN CENTRE

Thursday, 26th February, 6.45 p.m. Annual General Meeting followed by an Ordinary General Meeting in the Grand Hotel, Bristol.

The following meetings will be held during March:—

LONDON

Tuesday, 10th March, 5.30 p.m. General Meeting at Storey's Gate, St. James's Park, S.W.1. Paper: "Engineering Changes," by Harold Drew, B.Sc., M.I.Mech.E.

COVENTRY CENTRE

Tuesday, 3rd March, 7.15 p.m. Annual General Meeting followed by Ordinary General Meeting in the Craven Arms Hotel, High Street. Paper: "Gas Turbine Development," by B. E. G. Forsling.

DERBY CENTRE

Monday, 9th March, 7.15 p.m. General Meeting in the Midland Hotel. Address by the Chairman of the Automobile Division, Mr. Maurice Platt, M.Sc.(Sheffield), M.I.Mech.E., entitled "The Changing Practice of Automobile Engineering."

LUTON CENTRE

Monday, 9th March, 7.30 p.m. General Meeting in the Town Hall, Assembly Rooms. Paper: "Some Problems Arising from the Wider Use of Small Diesel Engines," by J. H. Pitchford, M.A. (Cantab), M.I.Mech.E.

Plastics Exhibition

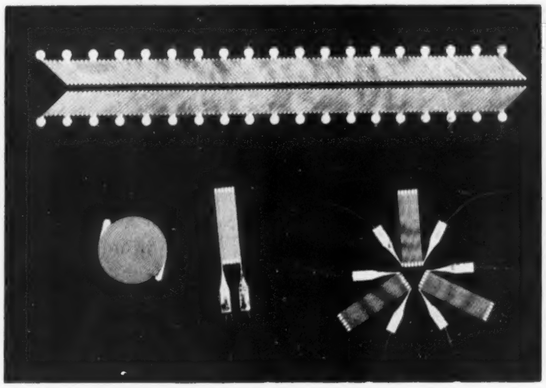
ON account of the Coronation, the opening of the second British Plastics Exhibition has been postponed from Wednesday, June 3rd, until Monday, June 8th, 1953. The closing date will be Thursday, June 18th. This exhibition, which is organized by our associated journal, *British Plastics*, will cover every aspect of the plastics industry. A Convention will be held during the period of the Exhibition. Technical papers covering a wide range of subjects of interest to plastics manufacturers and users will be presented at every session of the Convention.

FOIL STRAIN GAUGES

Some Recent Developments in Electric Strain Gauges

THE electric resistance strain gauge has been in use for more than ten years, and is such a simple piece of apparatus that at first sight there would hardly appear to be any possibility of its being improved. Nevertheless, Saunders-Roe Ltd., of East Cowes, Isle of Wight, working in conjunction with Technograph Printed Circuits Ltd., of London, W.1, have recently made marked advances in the development of this type of gauge.

In the conventional strain gauges a zig-zag pattern of fine resistance wire, the diameter of which rarely exceeds one thousandth of an inch, is mounted between strips of paper. This unit is cemented to the structure to be tested, and considerable care often has to be taken to waterproof it. The gauge extends with the test piece when loaded, and not only does it increase in length, but also the diameter is reduced. As a result, the electrical resistance of the wire increases in proportion to the dimensional changes. A suitable apparatus is employed to measure the change in resistance, and it may be calibrated so as to indicate the strain.



Some examples of foil strain gauges. Top: torque gauge; left: double spiral type; centre: linear type; right: rosette gauge

A fairly high degree of skill is called for in the production of the wire gauges. The wire is either wound on, and then cemented to, a thin backing paper, or wound on to a paper former to make a coil which is then flattened and the whole assembly is cemented to a paper backing. Finally, the ends of the wire have to be soldered or spot welded to relatively heavy lead-out wires or soldering tags, to which the leads from the electrical circuit of the strain measuring apparatus are joined. Then the gauge is reinforced with one or more strips of paper to furnish it with a reasonable degree of mechanical stability.

A much simpler method of production is employed with the new gauges. They are made from metal foil slotted alternately from each end so as to form a continuous zig-zag or looped strip which, apart from its rectangular section, is of much the same form as the normal circular section wire arrangement. The pattern is first printed on the foil, the ink used being acid resistant. Then the unprinted portion is etched away. Experimental patterns may be produced by a Technograph method by which the expense of preparing printing blocks is avoided, and the resultant patterns do not differ in any respect from those made in bulk production. In both experimental and normal production, the etching process is carefully controlled so that the degree of undercut is restricted. The final process is the cleansing of the product to ensure neutralization of all etch residues.

At the end of each loop, where the strip turns through 180 deg, the effective width of the foil is approximately $\frac{3}{64}$ in, so that the cross section is relatively large, and therefore has a low resistance. This reduces any inaccuracies which might possibly arise from end resistance effects. These large ends also give rise to more efficient transmission of the strain of the specimen to the element. Where the two leads of the measuring instrument circuit are attached, the width of the strip of foil is increased to about $\frac{1}{8}$ in, so that the leads may be soldered directly on to it. This arrangement eliminates the possibility of failure due to fractures which, on wire gauges, sometimes occur at the junction between the resistance wire and the lead-out wires.

Another advantage of the new gauges is that instead of mounting them in paper, they are mounted in very thin lacquer film, having excellent mechanical and electrical properties. This gives the foil element robustness and flexibility appreciably better than those obtained with many wire gauges and, since there are no intervening layers of paper, the transmission of strain from the specimen to the gauge is effected more positively than with wire gauges. Moreover, the foil gauges are completely waterproofed, with the result that the application for field tests is simplified.

Saunders-Roe Ltd. employ Araldite adhesive for mounting the gauge on the specimen. This is a thermo-setting plastic cement, which is cured at a temperature of about 100 deg C applied for 20 minutes. The heat may be applied either externally or by passing a current of from about 150 to 350 milli-amps for the required period of time through the gauge element itself. The current used depends, of course, upon the bulk and thermal conductivity of the specimen

Possibly the most important advantage of the foil gauge is that the ratio of contact surface area to the volume of the resistance element is relatively high, whereas in the wire gauge this ratio is a minimum, because of the circular cross section. As a result, the heat dissipation characteristics of the new gauge, and hence its current carrying capacity, is many times that of the wire element. The sensitivity of any resistance strain gauge is directly proportional to the current flowing through it. Thus, the increased current carrying capacity of the foil element represents a useful increase in sensitivity when the gauges are required to operate voltage sensitive recorders or indicators. Furthermore, because the power output of a gauge is proportional to the square of the current input, considerable advantage is to be gained by using the foil element to operate power sensitive apparatus. As much as 36 volts have been applied to these elements, but an e.m.f. of 12 volts is usually employed by Saunders-Roe.

The method of production of these gauges is so flexible that any conceivable 2-dimensional pattern may be produced with ease. This flexibility has led to the development of two variants of the common pattern of gauge element. The first is the torque gauge. In this variant the slots in the strip of foil, instead of being arranged so that they are open alternately at each end, are etched alternately from each side at 45 deg to the major axes of the strip. Two strips are mounted side by side around the periphery of the test piece, with the slots on one set at an angle of 90 deg to the slots on the other. This arrangement takes the place of the two elements, normally employed when a conventional wire gauge is used, and it balances out any longitudinal stresses superimposed on the torque. These stresses balance out because the gauges form opposite arms of a Wheatstone bridge in the measuring instrument. Torsional stresses are equal and opposite in each gauge and therefore their effects are added.

There are three advantages, in addition to those already mentioned, to be gained by using foil gauges for torsional strain measurement. First, alignment

of the gauges at 45 deg is a much simpler task because they have only to be wrapped round the shaft, and no marking out is necessary. Secondly, on running shafts, where a slip ring pick-up has to be employed, the increased output of the foil type element is useful. The third advantage is that the strains in the shaft may be integrated circumferentially, and not picked up locally as is the case with a wire gauge.

A different application of the foil principle is the diaphragm gauge. This is designed for use on relatively stiff diaphragms with small total deflections, since under these conditions the strain behaviour is not too complex. In this form, the gauge may be used to measure pressure, either static or pulsating, as well as strains. A double-spiral pattern is employed, the spiral being wound inwardly towards the centre and then doubling back on itself to the periphery again. This pattern integrates the circumferential strain over the area of the diaphragm, and thus makes the most efficient use of the gauge material and the strain response of the diaphragm. It also has the other advantages already mentioned as being characteristic of the foil type gauges.

Another design is being developed for very flexible diaphragms. It consists of two parallel linear patterns in the central tensile zone, coupled with two radial patterns in the outer compressive zones. The whole configuration is to be used as a four-gauge bridge of high power sensitivity. For a particular application of the diaphragm gauge, Technograph Printed Circuits Ltd. have been asked to devise a means of making the connections to the gauges in a very restricted space. This has been done by designing a printed and etched conductor pattern of copper foil, which is laid over the gauge pattern and soldered at the appropriate points.

Four applications of this type of gauge have been described. However, there must be many more which could be devised to suit special requirements. Because of the simplicity of the manufacturing process and the ease with which different patterns may be produced, there would seem to be little doubt that further development of the foil type of gauge will be made in the near future.

M.I.R.A. REPORTS

Recent Releases for General Circulation

AMONG the reports that have been issued by the Motor Industry Research Association, the following have recently been released for general circulation. Report 1950/10, *An Investigation of the Stresses in P.S.V. Type Wheels*, is the first of two reports dealing with this subject. It is confined to stresses due to static loading only. Results are given of stress measurements obtained with wire resistance strain gauges on P.S.V. type wheels. Stresses, due to tyre inflation and vertical static loading were measured, and when combined, these may be regarded as equivalent to the stresses to which the wheels are subjected during straight running on a smooth surface. The work was carried out on three- and four-piece wheels. Each type was tested with open valve slots incorporated, and again with the valve slots bridged. The effects of off-set, method of mounting, and different tyre sizes were also studied. The sizes of the wheels used were 5.00 × 20, 6.00 × 20 and 7.33 × 20.

The highest stresses measured occurred at points corresponding to those at which failures have been experienced in service, mainly at the fillet of the fixed flange of three-piece wheels and in the gutter. Stresses due to inflation were substantially proportional to tyre pressure, and load stresses were proportional to the applied load up to the rated load of the tyre. On three-piece wheels, the larger the wheel the lower were the stresses, while for a range of wheels of a given size, the stresses were similar regardless of who were the manufacturers. Oversize tyres usually gave higher inflation stresses than the recommended ones, and at the rated loads, they gave load stresses approximately proportional to the loads.

On four-piece wheels the load stresses were low. On the other hand, both high and low inflation stresses were encountered on these wheels depending on the fit between the loose flange and the retaining lip. However, from the data available, four-piece wheels appear to be superior to the three-piece ones. In the gutter, the variation of stress was similar in both the four- and three-piece wheels, and was attributed to high spots on the mating parts. High compressive stresses were measured at the open ends of the valve slots, but where closed slots were employed, these values were less than half those obtained with open ones. On the disc, the inflation stresses were negligible and the load stresses small. However, in cases where the discs were not flat, fairly high stresses occurred when the wheel was bolted to the hub.

Report 1951/9, *An Investigation of the Stresses in P.S.V. Type Wheels*, is the second on this subject, and it deals with the stresses due to dynamic loading. However, it includes appendices giving the results of further static tests relevant to the analysis of the dynamic tests and of measurements on the loose flange, which were not included in the original programme. Some stress measurements were also taken on a welded-disc wheel.

A double-deck P.S.V. was used for the tests. Space considerations limited the range of wheel sizes to 5.00 × 20 and 6.00 × 20, but within this range the stresses induced by such manœuvres as bumping, cornering and braking were found to be relatively independent of both tyre and wheel size. Both distribution and magnitude of the stresses obtained during straight running were similar to those observed in the static tests, but the rim stresses at 20 m.p.h. were slightly higher than the static stresses at corresponding wheel loads.

Cornering stresses appear, under these test conditions, to be more serious than either bumping or braking. When cornering at 0·42 g true radial acceleration the stress increases, by comparison with normal running stresses, were 100 per cent on the front wheel furthest from the centre of turn, and 55 per cent on the rear. On the disc, increases of several hundred per cent were found and, with 1½ tons normal load on a 5·375 in offset wheel, the maximum stress range was 30 tons/in^2 on the inside of the handhole. Under 0·5 g deceleration or braking conditions, the rim stresses were not greater than those to be expected from normal load transference, and the stresses due to torque were negligible. On the disc, the maximum stresses were about half those obtained when cornering. Bumps applied only local stresses, whereas cornering subjected the whole wheel to several complete cycles of high stress. The stress increases induced by bumps were of the order of 100 per cent on the rim and 160 per cent on the disc. Other effects, including those of unequal pressure in the tyres of twin wheels, weaving, glancing blows due to hitting the kerb, and a slow run on the proving ground pavé were also studied.

Laminated Torsion Bars

IN the description of the Lanchester Leda chassis published in the January 1953 issue of this journal, particular attention was drawn to the front suspension system in which laminated torsion bars are incorporated. Unfortunately, it was not made clear that the development of the laminated torsion bar was pioneered by Geo. Salter and Co. Ltd., West Bromwich, nor that the results quoted in the article were obtained through consultations between that company and the designers of the chassis.

ENDURANCE TESTING

An Important Development by Morris Motors Ltd.

THE 10,000 miles endurance test on a Morris Minor vehicle recently carried out on the Goodwood race circuit is without parallel in the automobile industry. Non-stop endurance bench tests of engines are, of course, common, but for this test it was decided that the vehicle should run for 10,000 miles under its own power for the whole time. This decision raised problems that were wholly novel. Hitherto, endurance tests always included stops for servicing and changes of crews, but through the use of a research tender of unique design, the Morris test was completely non-stop.

Essentially the aim was to carry out what was in effect a road test under much more closely controlled conditions than had ever before been attempted. Inevitably there must be some uncontrolled variables in such a test, but provision was made for recording these variables so that at the end of the test there would be a possibility of correlating the effects of variables on performance.

The research tender

In effect, the research tender was designed to act as both a mobile dynamometer and a service station. Certain of the servicing was relatively simple, but considerable ingenuity was necessary to develop means for changing wheels, and particularly either of the rear wheels, while the vehicle was still proceeding under its own power. All these problems were successfully overcome.

A tractor and a cab or control section comprised the research tender. The tractor was built from a Cowley truck, and incorporated a semi-universal articulation coupling contrived by mounting a road wheel and hub assembly to give radial movement about the stub axle and vertical articulation by the flexing of the tyre.

The controls section was fabricated from standard rolled mild steel sections of 3 in × 1½ in channel, 1½ in × 1½ in angle iron 1 in × 1 in T-section. These materials were arranged to give a minimum margin of safety of 7 : 1 when 10 persons and the jacked load of the front end of the test vehicle were being carried. The longitudinal members which acted as platforms from which vehicle servicing was carried out were fabricated from rolled sheet, with a 14 s.w.g. top skin, 20 s.w.g. lower skin, flanged boxing ribs of 16 s.w.g. spaced 12 in apart, and 14 s.w.g. vertical sides.

There were five separate systems for the trailer suspension. They were:—

(a) The flexible construction of the tender assembly.
(b) The rear suspension of the tractor and pivot wheel tyre.
(c) The 9 in × 5¼ in rear aero-type tyres.
(d) Twin cantilever springs with piston dampers at each rear wheel.
(e) Trailing castor wheels at the inner base of the fork, positioned to make ground contact during cornering under extreme loads.

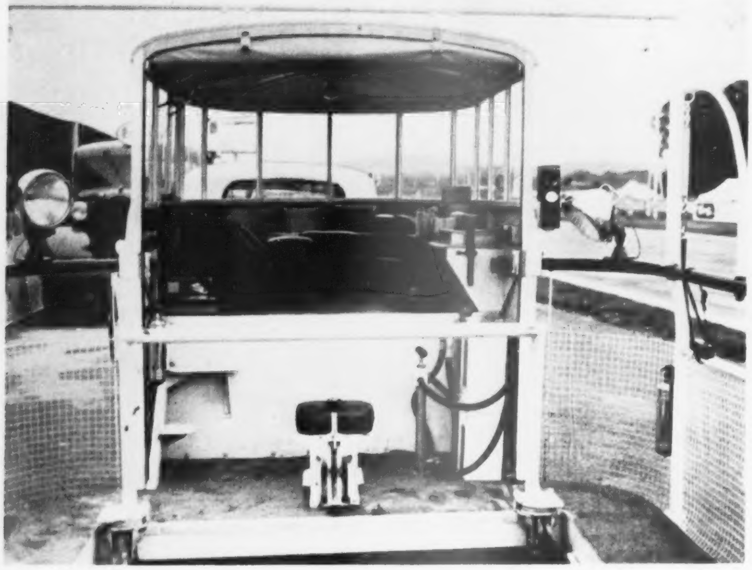
Smooth, vibration-free negotiations of road obstacles was essential. It was obtained by employing the following stiffnesses for the five spring systems:—

1. The construction of the tender assembly is such that flexibility of the sides of the fork in a vertical bend direction is 1000 lb/in.
2. The rear suspension of the tractor and pivot wheel has a vertical stiffness of 280 lb/in. The roll flexibility, which is mainly a function of the tractor rear suspension pivot wheel and fork sides flexibilities, is equivalent to 1 deg roll deflection for 294 ft/lb roll torque.
3. The rear aero tyre has 1400 lb/in vertical stiffness.
4. The vertical stiffness of the twin cantilever rear springs is 485 lb/in.
5. The trailing castor wheels provide trailer suspension stops and have no appreciable flexibility, that is, they may be considered as of infinite stiffness.

As the tender was to be used as a mobile service station the following equipment was installed either in or on the tender:—

1. The hitch for coupling the test vehicle to the tender.
2. A Panhard roof for positioning the car rear in the fork.
3. Hoisting equipment.
4. Double acting syringe pump with

The Morris research tender



The operator's control room on the tender

two containers in circuit for oil changes.

5. Swinging jigs for wheel changes.
6. Refuelling cans.
7. Stroboscope with generator and storage batteries.
8. General illumination equipment.
9. Address amplifier between tender cab and tractor cab.
10. Servicing tools and requisites.
11. V.H.F. radio telephone set.

The hitch

When for any reason the test vehicle was driven in to the fork of the research tender, the two vehicles were coupled together by means of the hitch, the front bumper of the test vehicle having a pegging plate for hitching to the tender. The hitch yoke is arranged to give considerable vertical freedom to the eye engagement of the car. This latitude is necessary to cater for two extreme conditions, (a) when the test vehicle is driven in with a flat front tyre, and (b) when the vehicle is hoisted to allow a front tyre to be changed.

The hitch drawbar proper is instrumented to record clearly both the magnitude and direction of the force imposed on it. An annular rack rotates a gear-driven solenoid and a calibrated reading of the force in lb is given on a gauge in the cab of the tender. A roller on a cam chamfer trips a microswitch series to indicate the direction of the force. A red light indicates that the tender is towing the test vehicle, an amber light indicates coasting with no force either way, and a green light signifies that the car is exerting a thrust against the tender. This light set is so positioned that it can be seen by the driver of the test vehicle so that he can carry out the necessary throttle manipulation to cause the green light to show. The drawbar is mounted between equi-rated springs that serve two functions:—

(a) The cushioning of the engagement force.

(b) The force displacement calibration for both the gauge and light indicators.

The Panhard rod

The Panhard rod for positioning the car in the rear of the fork is a constant length hook pivoted off the near side of the fork bridge tie. It is engaged in a hole in a bracket carried on the rear bumper of the test car. This bracket is perforated in several places so that the car can be centralized in the fork or offset within the fork, depending upon the nature of the work that is to be carried out.

Hoisting equipment

In determining the methods to be employed for tyre changes, consideration was given to both jacking and hoisting, the latter being chosen because it would give greater accuracy for measuring the thrust force through the hitch. The hoisting equipment consists of a chain block suspended from the cross-over beams of the tender structure at the required location to allow the corner of the test vehicle to be lifted.

Double acting syringe pump

There are two containers in circuit with the syringe pump, one for clean oil and one for used oil. For neat stowage the tins are rectangular in section. They are capable of receiving or discharging four gallons. A sighting window is cut in the face of each tin to indicate levels, and since the suction pipe of the pump draws oil from the sump at the correct level, the change in sighted level between used and fresh oil is a direct measure of oil consumed. The pump is $1\frac{1}{2}$ in diameter $\times$ 12 in stroke. It draws oil at $\frac{7}{8}$ pints per stroke and discharges at the same rate.

Swinging jibs

Swinging jibs are fixed into pivot sockets on the trailer side tubes adjacent to the centre lines of front and rear axles. Each job is cranked to swing almost to the road wheel side. For removing a road wheel for a tyre change, a tension spring with a hook on its lower end is attached to the appropriate jib. This spring is sufficiently strong to recoil with the weight of the road wheel, thus preventing the wheel from making contact with the ground after it was unbolted.

Refuelling cans

The design of the refuelling cans was developed from a five gallon proprietary can, which was chosen because of its neat stowage shape. The can was modified to include an air vent pipe, which was plugged when not in use. This pipe was laid from the atmosphere to the highest point of the tank tilted to supply angle. A screw was fitted on the feed nozzle and lifting handles were fitted at the appropriate places. The cans were stowed on the fork sides adjacent to the rear of the test car. They delivered

Test car entering the tender for a crew change

at the rate of five gallons in 30 seconds.

The test vehicle

The test vehicle was a standard Morris Minor saloon with B.M.C. overhead valve engine, that was standard in every respect except for adaptations that were necessary for the purpose of the test. These adaptations in no way affect the validity of the test results as applied to a standard production vehicle. The adaptations were:—

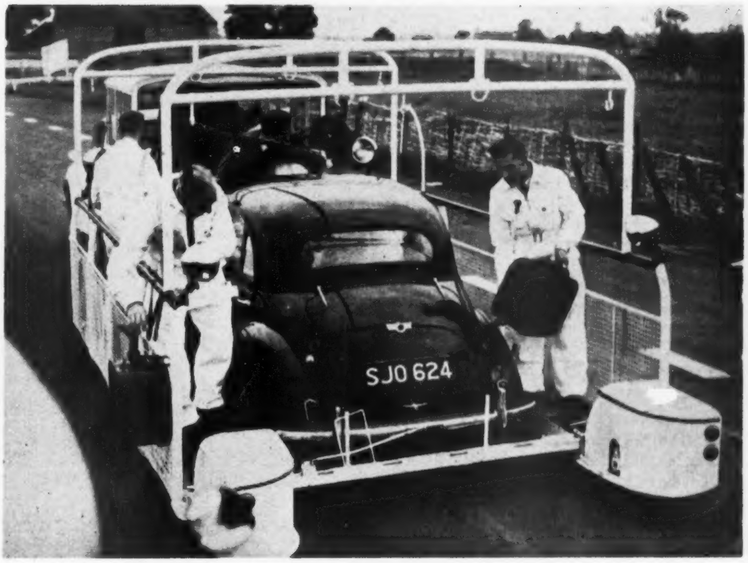
Power unit, cooling jacket and radiator system

An 85 deg C thermostat was fitted to compensate for the non-use of full power output on the test circuit. Bulb thermometers were fitted to record jacket inlet and outlet temperatures, and adaptors were fitted to the radiator top and bottom tanks. A draw blind was fitted forward of the radiator film block. To minimize fan suction effect, the blind was being lowered with the vehicle at speed, vertical wires were fitted between the block and the blind. This was to facilitate temperature control at approximately two-thirds output at 3,500 r.p.m.

A float level indicator was fitted within the engine sump to record oil level. This unit gave quantity readings on a gauge fitted on the test instrument panel. Temperature recordings of crankcase and gearbox oils were effected by means of bulb thermometer adaptors fitted into the components. The power unit was run-in on a test bed for 20 hours at full load at 3,500 r.p.m.

Final drive assembly

As a precautionary measure and to ensure freedom from any danger of seizure when rear wheels were being changed, the final drive assembly was motored through the differential for one hour at approximately 10 m.p.h.



Refuelling The test car exerts a positive thrust of at least 10 lb on the tender

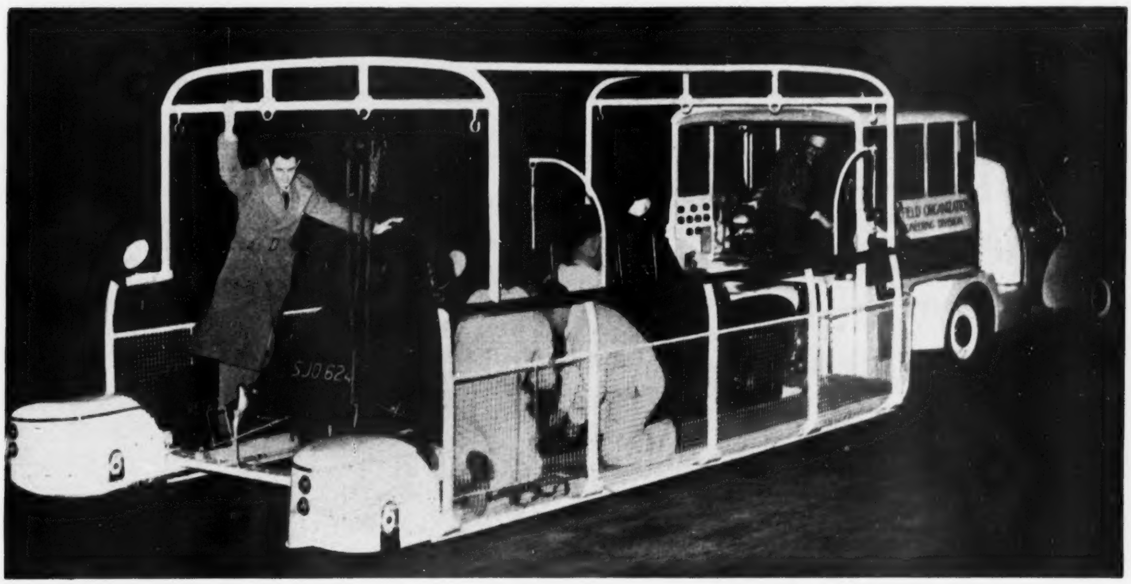
To facilitate oil temperature measurement, a bulb adaptor was fitted to the axle. Special arrangements had to be made to allow rear wheel changes to be made while the vehicle was travelling under its own power. First, it was necessary to incorporate means for locking the wheel that was to be changed. This was effected by fitting a large knurled adjusting nut on each rear hand brake cable so that there was independent locking for each wheel. In addition, each half of the rear axle had a wire rope cable attached to it. This cable passed through the body floor into the boot where it passed into a split metal tube. This arrangement locked the rear suspension and prevented the axle rebounding away from the body on to the springs when the body was raised for wheel changing.

Front suspension

Blocks were supplied for insertion between the front shock absorber and the bottom side of the wing valance aperture. These were also adopted to interrupt suspension rebound when a front corner of the car was being hoisted for a wheel change.

Road wheels, assembled

All wheel discs were perforated and grommeted to provide timing revolution pick up for stroboscopically examining tyres for tread wear. All the tyres were prepared and assembled on to the road wheels in good static and dynamic balance by the Dunlop Rubber Co. Ltd. The tyre tread form was normal



Servicing the test vehicle at night

export standard, that is 9 mm tread, 6 ply. Certain tyre modifications were carried out as a result of preliminary tests. These were: —

(a) Harder rubber to give high cornering power.
(b) White marker strip insert immediately next to the ply to give final warning that wheel change was necessary.
(c) A rubber lining inside the case next to the tube to give protection against puncturing.
(d) The tubes were also puncture proofed by internal sealing.

Electrical system

The radio telephone equipment imposed an additional three amp load on the electrical system. It was extremely likely that this additional drain on the battery would occur continually during night fog conditions, when all lights and windscreen wipers would also be in service, together with panel and auxiliary panel illumination. Therefore, a standby battery was installed behind the rear seat squab with a cross-over switch above the parcel tray. An auxiliary coil was also fitted. This, together with the service coil, was positioned above the parcel tray, with a cross-over low tension switch. In case of emergency the intention was to switch the high tension lead as quickly as possible. An auxiliary fog lamp was fitted on the off-side of the bumper to direct the light beam to the right, since the test was to be run on a closed left-hand circuit.

Fuel system

A 12 gallon tank was installed in the standard position. The tank section was current standard, and the upper half was extended to protrude in to the spare wheel compartment. This was possible because a spare wheel was not carried. Two fuel pumps were fitted.

Driving compartment

To facilitate change-over between driver and observer, two standard Minor bucket seats were modified to have pivoting squabs which contacted the rear seat cushion. A support strap, clipped to a ring on the left-hand " B " post, was supplied to give the observer body support on continual left-hand bends.

Lubrication system

To facilitate oil changes, inlet and outlet pipes were laid to the sump, gearbox and final drive. The pipes for the sump and gearbox ended at the off-side front valance beneath the bonnet for coupling to feed and drain hoses from the tender installation. The inlet and outlet pipes from the final drive were taken to an installation behind the rear seat squab, comprising a double acting syringe pump with a fresh and used oil container in circuit. This pump was accessible from inside the car.

Sources of general thermal interest

Thermo-couples were installed at several locations. They all had leads taken to a junction box fitted beneath the fascia adjacent to the " A " post. A press button switch gave contact with a Cambridge temperature recorder within the glove box, but arranged to slide out into vision line. The thermo-couple locations were: —

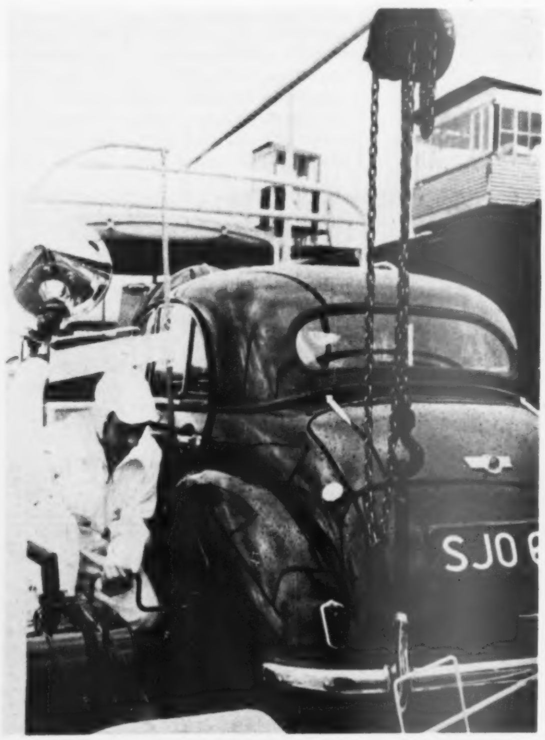
Battery electrolyte.
Off-side front shock absorber.
Both rear shock absorbers.
Fuel float bowl.
Under-bonnet atmosphere.
Fuel pump.
Car interior, at roof centre on the floor at passengers' feet.
The dynamo. (This thermo-couple was not an original fitting, but in the early stages of the test showed desirability for knowing the generator temperature variations with charge rate fluctuations.)

Radio telephone equipment

V.H.F. communication equipment was fitted in the test vehicle. The power pack was installed beneath the observer's seat with the telephone ear-piece set at his left-hand side on the car floor between the seat and the sill. The aerial cable was laid up the near side centre post to a small aerial mast in the centre of the roof.

The test circuit

Obviously a test of this character could not be carried out on the road, and after consideration of the possible venues the Goodwood circuit was chosen. Even though this is a closed track it is not circular, nor is it absolutely flat. It therefore simulates road conditions reasonably well. It was



Removing a rear wheel. The other rear wheel is exerting a positive thrust on the tender

The rear wheel removed from the car but still suspended from the swing jib

decided to carry out the test in the anti-clockwise direction. The circuit therefore comprised a 90 deg left-hand bend (Woodcote), a 150 deg left-hand bend (Lavant), followed by a rising 80 deg right-hand bend, the remainder of the circuit being a progressive bend amounting to 200 deg. The actual road length used by the car when cutting the corners was 2·353 miles per lap as recorded by instruments on the car, but for calculations the B.R.M.C. figure of 2·38 miles was used.

It was realized that the test would be extremely severe on tyres, and a series of preliminary tests were carried out to establish safety margins. To begin with, the rough "Resmat" surface of Goodwood is much more abrasive than the average road surface. Secondly, to maintain a mean speed of 45 m.p.h. the centripetal acceleration at the bends varied from 0·25 to 0·4 g, and this in turn was reacted by a vehicle-to-course coefficient that varied between 0·6 and 0·57.

From measurement of the petrol consumed at 45 m.p.h. on a straight course and comparing this with the consumption during test runs on the circuit, it was found that the effort required to overcome centrifugal forces as reacted by the road wheels and tyres was equivalent to using more fuel at the rate of x – 7·62 miles per gallon where x = the consumption on a straight course. As the drag horse power at 45 m.p.h. is known to be approximately 13 h.p., a figure of 1·7 h.p. can be equated from the increase in consumption. This therefore is the h.p. required to negotiate the course itself.

It is interesting to note that the extra power required merely for negotiating the circuit would, over the 10,000 miles of the test, be equal to the energy needed for climbing approximately 66 miles. In other words, the difference between 10,000 miles of normal roads and 10,000 miles of the Goodwood circuit is that the latter requires 807,800,000 ft-lb more energy and is equivalent to 15 per cent more miles.

Some interesting figures on tyre wear were obtained from the preliminary tests. For example, when the circuit was covered at the maximum speed within the capacity of the vehicle, that is, 59 m.p.h. mean lap speed, the wear rate on the front tyres was measured at 37·7 miles per 1 mm of tread depth, as compared with 250 miles per 1 mm of tread depth at a mean speed of 45 m.p.h. A test, run at full lock with a 15 ft radius and at the adhesion limit of 7-8 m.p.h. increased the tyre wear enormously to only four miles per millimetre of tread depth.

The instrument panel in the test vehicle. The press button junction box for bringing the thermo-couples into circuit is at the left

On normal A and B class roads, and at a mean speed of 45 m.p.h., the wear rate is in the order of 1500 miles per millimetre of tread depth at the off-side control wheel. These figures clearly show the severe conditions for the tyres on an enclosed small perimeter track. The additional horse-power required for changing direction is absorbed by removing rubber at six times the rate experienced in normal road operation.

The preliminary test runs proved beyond any doubt that if vehicle stability and operational safety were to be maintained throughout the test, several tyres would be necessary. It was therefore necessary to make provision for examining the tyres while the vehicle was running. Two methods were adopted. During the hours of darkness tread depths on the tyres were estimated to a reasonable degree by using a stroboscopic lamp adjusted to wheel rotation frequency.

Another method of judging tyre wear during daylight was to drive the vehicle past the control so that the tyre could be washed with a sponge. If the marker strip incorporated in the tyre had been reached, it was easily discernible. This method was used at the same time as the rear axle oil change since this also required a slow speed. Incidentally, the rate of tyre wear was remarkably consistent on all the tyres used.

In running the 10,000 miles at an average mean speed of 45 miles per hour, it was hoped to obtain 45 miles per gallon of fuel, 6,000 miles per gallon of lubricating oil, to use not more than three sets of tyres and to complete the 10,000 miles with the original sparking plugs. The final results showed very close approximations to these figures.

The test covered 10,000·76 miles in 221·089 running hours to give a mean speed of 45·32 m.p.h. Consumptions for the test were:—

Fuel, 232·59 gallons,	43 m.p.g.
Engine oil, 9·2 pints,	8600 m.p.g.
Radiator water, 11·75 pints,	6800 m.p.g.
Tyres	10

Remarkably little trouble was experienced with the equipment although it was found necessary to change one petrol pump and a voltage regulator.

Although the project was referred to as an endurance test, it was hoped to obtain useful information on the effect of atmospheric conditions on temperatures. Good fortune attended the project in this respect, since during the period there were wide variations in the air ambient temperature and in humidity. The maxima and minima for atmospheric conditions were:-

Air ambient temperature	18 deg C	2·5 deg C
Barometer	30·3 in Hg	29·15 in Hg
Humidity	100 per cent	44 per cent

At the moment it is not possible to discuss the information obtained from the test, but reference may be made to one or two interesting points. For example, it was clearly shown that the test vehicle ran for more than 4,000 miles before running-in was completed. This was evidenced by a gradual reduction in fuel consumption until finally the consumption was virtually constant for the remainder of the test. Before the test started, acceleration and maximum speed tests were carried out, and at the conclusion of the test they were repeated at Goodwood and Cowley. There was no significant difference in the results of all three tests.

In every respect, the test was remarkably successful. That the vehicle would complete a non-stop run of 10,000 miles with only minor failures of ancillary equipment is gratifying but not surprising. Probably the most valuable results of the test lie in the accurate information that has been obtained about temperatures under widely varying atmospheric conditions. It is, we believe, the first time that this problem has been studied systematically and scientifically. The experience gained with a research tender that acts as a mobile dynamometer is also of major importance, since it has created possibilities for other studies.

AUTOMOBILE DYNAMIC LOADS

Some Factors Applicable to Design

By T. K. Garrett, A.M.I.Mech.E., A.F.R.Ae.S.

In the past it has been customary to design a vehicle, produce a prototype, test it, and then strengthen it where necessary. This procedure can only be satisfactory if the test conditions correspond closely with those obtained in normal everyday use, and at the same time include the worst conditions for which the vehicle is to be designed. So far as strength is concerned, two separate factors influence design. They are: fatigue due to repeated loads, the magnitude of which may vary from light to moderately heavy, and the possibility of failure due to exceptionally heavy loads, such as may occur when unexpectedly encountering a large obstacle in the road. The problem is further complicated by the fact that conditions vary widely as between one country and another.

It follows, therefore, that the only sure way to test a prototype so as to prove that the design is one hundred per cent structurally sound, is to take it to those countries in which conditions are most severe. This would involve very extensive tests in countries where extremes are to be found of temperature, as well as of road surface, gradient, and traffic conditions. Such tests as a regular procedure are obviously out of the question, although in the past they have been resorted to in a few isolated cases.

Simplified tests

The problem is one of devising a satisfactory substitute for these comprehensive tests. The method usually employed is to accelerate failure by driving the vehicle for some hundreds of miles over extremely rough artificial surfaces or by mounting the wheels on rotating drums fitted with bump obstacles on the circumferences to apply loads at a comparatively high frequency. However, in all these tests there is a large element of uncertainty as to how they can be correlated with road conditions. Furthermore, it is always difficult to decide just how rough the test track should be, at what speed the car should be driven, and how many miles it should be expected to cover without showing signs of structural failure.

It is surprising that more comprehensive investigations have not been made to determine the magnitude of the loads for which a vehicle should be designed. This is probably an easier task than is generally realized. Although fatigue due to frequently repeated light loads must be taken into consideration, the evidence provided by practical experience seems to indicate that if a vehicle is designed to withstand the most severe loads to which it may on comparatively rare occasions be subjected, it will be more than strong enough from the point of view of fatigue. This is only true, of course, provided severe stress concentrations are avoided.

If the maximum loads to which the vehicle will be subjected are known, it is possible to design so that each component of its mechanism and structure is only just strong enough. This leads to economy in material. It also results in a considerable saving of time and money at the development stage because fewer modifications are needed to strengthen weak points, and a much simpler and quicker test procedure can be adopted. The simplification of the test procedure is possible since only the critical loads are applied; the structure that is strong enough to withstand these being more than adequate to cope with any complex combinations of lighter loads.

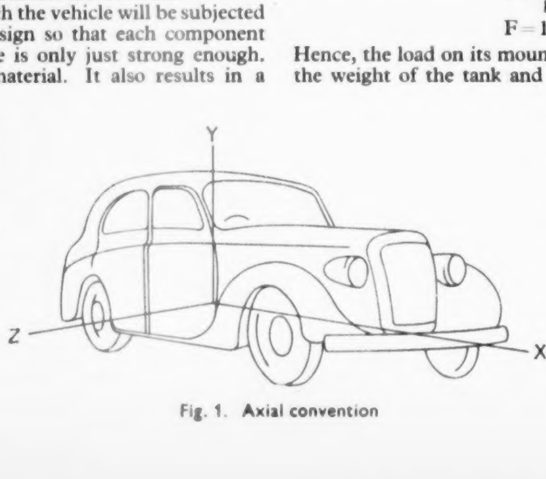


Fig. 1. Axial convention

In applying these principles to design and testing it is necessary first to determine exactly what these loads are. Of recent years too much emphasis has been placed on the determination of stresses in vehicle structures, without reference to the loads producing them. These stresses are, of course, only applicable to the particular vehicle being tested, and a single minor modification, such as the incorporation of a small weld at the base of a pillar, can completely alter the stress distribution throughout the whole of the structure. Experimental work, with strain gauges, to determine stresses in vehicle structures is, however, of value in an indirect way. From electric resistance gauge recordings of strains in components, first the stresses due to the static weight of the vehicle with its load are calculated. Next, the stresses under dynamic load conditions on the road are found in the same way. The ratio of these two stresses can then be used in the following manner as a multiplication factor for design purposes.

When initiating a new design, an estimate is first made of the weight, centre of gravity position, and wheel reactions of the vehicle with its load. A method of doing this was described in "Weight Control", in the October 1951 issue of the *Automobile Engineer*. From the wheel reactions, the static stresses can be calculated. Then, by using the multiplication factor, the magnitude of the dynamic stresses may be determined.

This determination is based on Newton's second and third laws of motion in which it is stated that acceleration is proportional to the applied force producing it, and that to every action there is an equal and opposite reaction. These laws demonstrate with mathematical precision exactly what loads are applied to a vehicle. If this is not apparently borne out in practice, the fault lies in the failure of the investigator to take all relevant factors into consideration.

It is most convenient to express dynamic loads in terms of acceleration since:

$$F = \frac{Wa}{g}$$

where F = the applied force
W = weight of the component
a = acceleration of the component
g = acceleration due to gravity.

Furthermore, if the acceleration is expressed in terms of g the applied force can be expressed in terms of the weight of the component. Thus, if the petrol tank in a car is moved upwards with an acceleration of 48 f/s, which is equivalent to $1\frac{1}{2}$ g acceleration, it follows from the formula that

$$F = \frac{W}{g} \times 1\frac{1}{2}\,g$$

$$F = 1\frac{1}{2}W$$

Hence, the load on its mounting bolts or brackets is $1\frac{1}{2}$ times the weight of the tank and its contents.

Vertical, lateral, and fore-and-aft loads

Some work has already been carried out to determine the magnitude of the dynamic loads experienced in automobile structures. In a paper entitled *Stress Engineering as Applied to Automobile Bodies*, (S.A.E. Quarterly Transactions, July 1948) Johnson and Heyl state that the greatest vertical acceleration they have obtained in tests is 3 g. They also state that

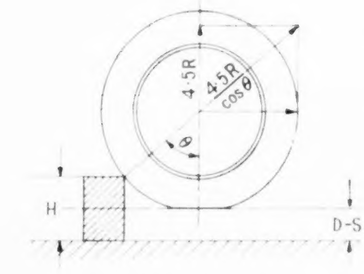


Fig. 2. Diagram of the vertical and-fore-and aft loads due to a bump

the maximum torque applied to a vehicle occurs when it is supported on three wheels in a static condition. Although experience suggests that the vertical acceleration of 3 g is correct, many would not agree with their statement concerning the maximum torque. One wheel can quickly rise to the full bump position and impose heavy dynamic loads on the structure before the whole vehicle has had time to rise sufficiently for the static three wheel support condition to be reached. It is probable, therefore, that the maximum torsion that will be applied to the structure is very nearly equal to that applied by an acceleration of 3 g on one wheel. This load will be reacted by the inertia of the sprung mass.

Experimental work, on commercial vehicles with strain gauges, in this country, has suggested that rebound loads are at least as great as, if not greater than, bump loads, although it is felt that more research is needed on this subject. The reason for such high rebound loads is rather obscure, but presumably they could be caused by the sudden arresting of the unsprung masses on their rebound stops, particularly in vehicles not fitted with independent suspension. Until more information is available it is deemed advisable to consider the vertical acceleration of 3 g to be reversible, particularly for the design of suspension components, and their adjacent supporting structure.

Fore-and-aft loads are caused by braking and accelerating. Crash loads can be ignored, since the less serious ones are taken by local denting of body panels, and no one, except perhaps a designer of armoured fighting vehicles, designs a structure to take shocks of the magnitude of those associated with a crash. The design of bumper attachments is at present a matter of guesswork, although it need not remain so if some research were to be done on the subject. Road tests show that the maximum braking retardation obtainable in passenger cars is approximately 1 g. It is somewhat uncertain as to what acceleration can be imparted to a car by suddenly letting in the clutch, but it is thought advisable to consider the braking load as being reversible, particularly in view of the fact that brakes can be suddenly applied when reversing. The magnitude of the reverse load again depends on the type of brakes used as, in reverse, some are more effective than others. It should be remembered that extreme retardation due to braking does not necessarily imply braking from high speed.

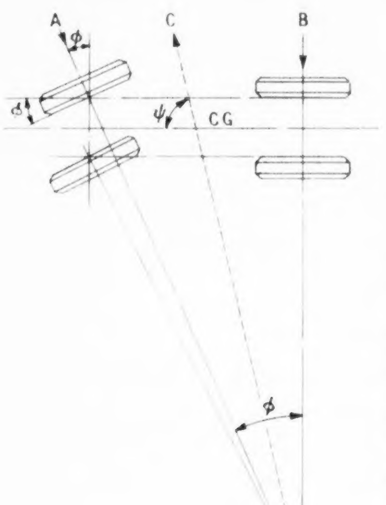


Fig. 3. Cornering force diagram for a vehicle

Finally, there are centrifugal loads due to cornering. In view of the fact that the wheels are on the point of locking at 1 g retardation during braking, it would seem reasonable to assume that the same loads will be applied in a lateral direction when the wheels are on the point of skidding while cornering. This does not necessarily follow however, since the cornering power of a tyre is not necessarily the same as its braking power. On the other hand experience does suggest that this figure of 1 g is, in general, reasonably near the truth. It is dependent among other things on the height of the centre of gravity of the vehicle since a vehicle with a high centre of gravity can overturn before it reaches the point of skidding. Here again, it is a simple matter to use an accelerometer to determine just what lateral accelerations are obtained.

In connection with loads due to cornering, braking and acceleration, it should be remembered that they are each dependent on the limiting friction between the tyres and the road. It follows, therefore, that they cannot all be applied simultaneously at their maximum values. If a car is on the point of skidding at a corner, and either the brakes or the engine power is applied, then skidding will take place, and as a result the loads at the wheels are likely to be reduced since the sliding friction coefficient is less than the static one.

From the foregoing information a set of loads for which the vehicle must be designed can be specified. Whilst the greatest magnitude of each of these loads can be determined accurately by experiment, it is not possible to be so sure of the homogeneity of engineering materials, or the consistency of surface finish, and many other variable factors affecting the strength of mechanical and structural parts. In addition, some margin of safety is needed so that the structure shall be, without any doubt, strong enough to cope with the worst loads that it may encounter. Accordingly, a factor of safety of 1·5 should be used.

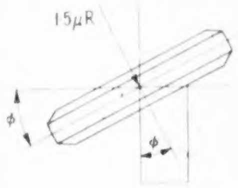


Fig. 4. Cornering force diagram for a wheel

The loads are then :

1. Vertical acceleration ± 4·5 g.
(hitting a bump)
2. Fore-and-aft acceleration ± 1·5 g.
(braking and accelerating)
3. Sideways acceleration ± 1·5 g.
(cornering L.H. or R.H.)

Doubt might be expressed concerning the validity of some of the acceleration factors quoted. They can, however, be checked by applying the appropriate loads to calculate the stresses in components of existing motor cars that have been proved by satisfactory service to be strong enough. Such a check will show that they each give resultant stresses conforming with current design practice as far as strength is concerned. In fact, there is at least one model in production that has been successfully designed, on the basis of calculations on these lines, by a large manufacturer in this country.

Braking and steering loads

Two further local loads must be considered. In the first place, there are the loads in steering mechanisms. These can only be obtained by experiment. It is possible, by strain gauges fitted to the steering rods, to determine the stresses and hence the loads in the rods under different conditions. These loads fall into two categories, those that are self-balanced through the track rod, and those that are balanced by the reaction of the steering box.

Very few results of tests to determine these loads have been published, but an indication of their magnitude can be given by the following. A car having a total front end reaction of 1,150 lb at the wheels was found to have a maximum out-of-balance load of 1,030 lb when hitting a kerb, in which condition the effective lever arm, about the swivel pin, of the load in the steering rod was about 4½ in. This gave a maximum torque about the pin of 4,640 lb-in.

A further interesting factor was brought to ligh in these

tests. Under certain conditions, when steering a straight course on pavé, an unbalanced load of 800 lb was experienced, which, with an effective lever arm of $5\frac{1}{2}$ in, gave a torque of 4,400 lb-in about the swivel pin. The surprisingly large magnitude of this load was probably due to a resonance in the steering system, giving rise to a vibratory motion of the wheels or of part of the system. This, being a condition likely to cause fatigue failures, should be investigated further.

Before the information concerning steering loads can be of general application, the effect should be determined of wheel size, offset of swivel pin from the centre of wheel contact area on the ground, swivel pin angle, camber and castor angle, and wheel vertical reactions. Then it is highly probable that a set of design curves can be drawn up, taking these variables into consideration. A load figure to be of universal application should be expressed as horizontal force ÷ wheel reaction, acting in the plane of the wheel and above the contact centre of the tyre with the ground. Loads in the steering mechanism could then be determined by calculation from its geometry. It may be found that the loads can be satisfactorily expressed for general application as a torque about the swivel pin, as has been done in the example already quoted.

The second of the local loads that must be considered is brake torque. Here again, experimental work is needed to determine the effect of brake hop, and to evolve a method of calculating the magnitude of the brake torque. One method that might be employed is to use a friction coefficient of unity—a value suggested by a braking retardation of 1 g—and multiply the wheel reaction by the rolling radius. In this case, the wheel reaction would be the sum of the static and weight transfer reactions.

Detailed load analysis

Before the loads already described can be correlated to give the most severe conditions for which the car must be designed, it is necessary to consider in detail what happens when a car hits a bump, and when it is steered on a circular course. The axial convention adopted is shown in Fig 1. When the wheel hits a bump with the brakes off, the resultant load can only act through the wheel axis, and if the factored vertical reaction is 4·5R, where R is the static wheel reaction, then the total load is $\frac{4{\cdot}5R}{\cos\theta}$ along a line through the wheel axis and the centre of contact of the wheel with the bump, where θ is the angle between this line and the vertical. From this it can be seen that the horizontal component is $\frac{4{\cdot}5R}{\cos\theta}\times\sin\theta = 4{\cdot}5R\tan\theta$

To determine the angle θ it is necessary first to decide for what bump to design. Assume this is a curb the height of which is H inches. Then, it must be remembered that the wheel will ride up over the curb and the maximum load will only be applied when the bump stop is struck. If the motion to full bump has been very rapid so that the body has only risen a negligible distance, the effective height of the curb is [H—(D—S)], where D = total deflection to full bump and S = static deflection, both dimensions being inches, Fig 2. A reasonable value for H would appear to be 6 in; this figure, of course, includes an allowance for tyre deflection.

Next to be examined in detail are the loads due to cornering. From Fig 3 it will be seen that the wheel reactions A + B must balance the centrifugal force C, acting through the centre of gravity. When the car is on the point of overturning there will, of course, be no reaction on the two wheels nearest the centre of turn. But even if the car is not on the point of overturning, the loads on these wheels will be much lighter than those on the outer ones.

The load due to cornering, on a wheel free to rotate, can only be perpendicular to the plane of the wheel, and is applied at the ground, Fig 4. Let R_A be the static reaction at the front wheels, and R_B the static reaction at the back wheels, then resolving the side load at the front into two components and applying the factor of 1·5, gives $1{\cdot}5\times\mu R_A\sin\phi$, along the x axis, and $1{\cdot}5\,\mu R_A\cos\phi$ along the z axis. It is the rearward component that will give rise to a certain amount of "roll-on".

R_A and R_B must be determined from the following considerations: centrifugal force F acting at the C.G. of the vehicle $= 1{\cdot}5\times\frac{Wv^2}{rg}$, where v is the velocity of the vehicle in ft/sec, W is the weight in lb, r is the radius in ft, and g is in ft/sec² units. This force can only be balanced by the vectorial sum of the horizontal reactions at the wheels, which is $\mu(R_A+R_B)$. Hence the largest possible value of the centrifugal force is μW.

Resolving the centrifugal force perpendicular and parallel to the x axis of the car, Fig 3, we have respectively $\mu W\sin\psi$ and $\mu W\cos\psi$. Then the unfactored reactions may be found from:

$$\mu(R_A\cos\phi+R_B) = \mu W\sin\psi \quad (1)$$

and

$$\mu R_A\sin\phi = \mu W\cos\psi \quad (2)$$

whence from (2):

$$R_A = W\frac{\cos\psi}{\sin\phi} \quad (3)$$

from (1) and (2):

$$R_A\cos\phi\sin\phi+R_B\sin\phi = W\sin\psi\sin\phi$$
$$R_A\cos\phi\sin\phi = W\cos\psi\cos\phi$$

Subtract:

$$R_B\sin\phi = W(\sin\psi\sin\phi-\cos\psi\cos\phi)$$

$$R_B = \frac{W(\sin\psi\sin\phi-\cos\psi\cos\phi)}{\sin\phi}$$

$$= \frac{-W\cos(\psi+\phi)}{\sin\phi} \quad (4)$$

The angles ϕ and ψ may, of course, be obtained from the layout of the vehicle, so that R_A and R_B may be determined respectively from (3) and (4). Since 180 deg $> \psi+\phi >$ 90 deg, the negative sign in the expression for R_B disappears.

The design cases

Having analyzed the loads into their individual components, and evaluated each, it is then necessary to synthesize them again in all their possible combinations. Each combination of loads will be called the *design case*. There may be several design cases: for instance, it is likely that the design case for the front suspension will be one that includes loads due to braking. On the other hand, the rear suspension is more likely to be designed by a case in which the car is accelerating. The following symbols will be used in addition to those already employed in connection with cornering loads.

Let W = laden weight of the vehicle

$\bar{y}$ = height of the centre of gravity of the laden vehicle above the road. (A method of determining $\bar{y}$ is given in an article entitled "Weight Control" in the October 1951 issue of the *Automobile Engineer*.)

B = the wheelbase

T = the track

R = the laden static reaction of the wheel under consideration

θ = effective angle of inclination of the line of action of the bump load from the vertical

Case 1. *Hitting a bump while braking*. This will be a design case for the front end of the vehicle. It might occur when the driver, seeing an unexpected obstacle or pothole in the road, applies his brakes hard and hits it.

(a) Vertical load due to bump

$$= R\times4{\cdot}5 \quad \text{lb/wheel}$$

(b) Rearward load due to bump

$$= R\times4{\cdot}5\times\frac{\sin\theta}{\cos\theta} \quad \text{lb/wheel}$$

(c) Roll-on load due to bump

$$= R\times4{\cdot}5\times\frac{\sin\theta}{\cos\theta}\times\frac{\bar{y}}{2B} \quad \text{lb/wheel}$$

(d) Roll-on load due to braking

$$= 1{\cdot}5\frac{W\bar{y}}{2B} \quad \text{lb/wheel}$$

(e) Rearward load due to braking

$$= 1{\cdot}5R+1{\cdot}5\times\frac{W\bar{y}}{2B} \quad \text{lb/wheel}$$

Vertical Component

$$= R\times4{\cdot}5 + R\times4{\cdot}5\times\frac{\sin\theta}{\cos\theta}\times\frac{\bar{y}}{2B} + 1{\cdot}5\,\frac{W\bar{y}}{2B}$$

$$= 1{\cdot}5\left\{3R\left(1+\frac{\bar{y}}{2B}\tan\theta\right)+\frac{W\bar{y}}{2B}\right\}$$

Rearward Component

$$= R\times4{\cdot}5\times\frac{\sin\theta}{\cos\theta}+1{\cdot}5R+1{\cdot}5\frac{W\bar{y}}{2B}$$

$$= 1{\cdot}5\left\{R(3\tan\theta+1)+\frac{W\bar{y}}{2B}\right\}$$

Lateral Component $= 0$

To these loads must be added the local effect, on the suspension components, of brake torque.

Case 2. *Cornering*. This case will probably give the worst side load conditions on the front end of the vehicle. It is impossible to apply an additional braking force when the wheels are already on the point of sliding in cornering. Moreover, if the front wheels were to hit a bump when set at an angle for a turn, there would be an outwardly directed component that would subtract from the inwardly directed centripetal force.

The coefficient of friction μ between the tyres and the road is taken to be unity. It is true that greater side loads than those derived from tyre adhesion could be obtained when hitting a curb during a high speed turn, but there is a limit to the maximum loads for which a vehicle can be designed. Even so, it is suggested that further research should be carried out on this aspect of vehicle dynamic loads.

Rearward component (which causes roll-on)

$$= 1{\cdot}5\,\mu W\cos\psi$$

Total vertical component $=$ weight $+$ roll-on

$$= 1{\cdot}5W\left(\frac{\cos\psi}{\sin\phi}+\mu\cos\psi\,\frac{\bar{y}}{2B}\right)$$

Lateral component $= 1{\cdot}5\,\mu R_A\cos\phi$

$$= 1{\cdot}5\,\mu W\,\frac{\cos\psi}{\tan\phi}$$

It will be noticed that because of the roll-on, the grip at the rear wheels will be reduced, and this will cause the rear end to slide before the full centrifugal load catered for by the assumption that $\mu=1$ is attained. This will be most noticeable during turns of a small radius. However, in view of the arbitrary nature of the assumption, it is hardly worth while to complicate matters by allowing for this tendency until more research has been done.

Case 3. *Hitting a bump whilst accelerating*. This will be a design case for the rear end of the car.

(a) Vertical load due to bump $= R\times4{\cdot}5$

(b) Rearward ,, ,, ,, ,, $= R\times4{\cdot}5\times\dfrac{\sin\theta}{\cos\theta}$

(c) Roll-on ,, ,, ,, ,, $= -R\times4{\cdot}5\times\dfrac{\sin\theta}{\cos\theta}\times\dfrac{\bar{y}}{2B}$

(d) ,, ,, ,, ,, ,, acceleration $= 1{\cdot}5\,\dfrac{W\bar{y}}{2B}$

(e) Forward ,, ,, ,, thrust at road $= -\dfrac{1{\cdot}5W}{2}$

Vertical component $= R\times4{\cdot}5+1{\cdot}5\dfrac{W\bar{y}}{2B}-R\times4{\cdot}5\times\dfrac{\sin\theta}{\cos\theta}\times\dfrac{\bar{y}}{2B}$

$$= 1{\cdot}5\left\{3R\left(1-\frac{\bar{y}}{2B}\tan\theta\right)+\frac{W\bar{y}}{2B}\right\}$$

Rearward component $= R\times4{\cdot}5\times\dfrac{\sin\theta}{\cos\theta}-\dfrac{1{\cdot}5W}{2}$

$$= 4{\cdot}5\,R\tan\theta - 0{\cdot}75W$$

Lateral component $= 0$

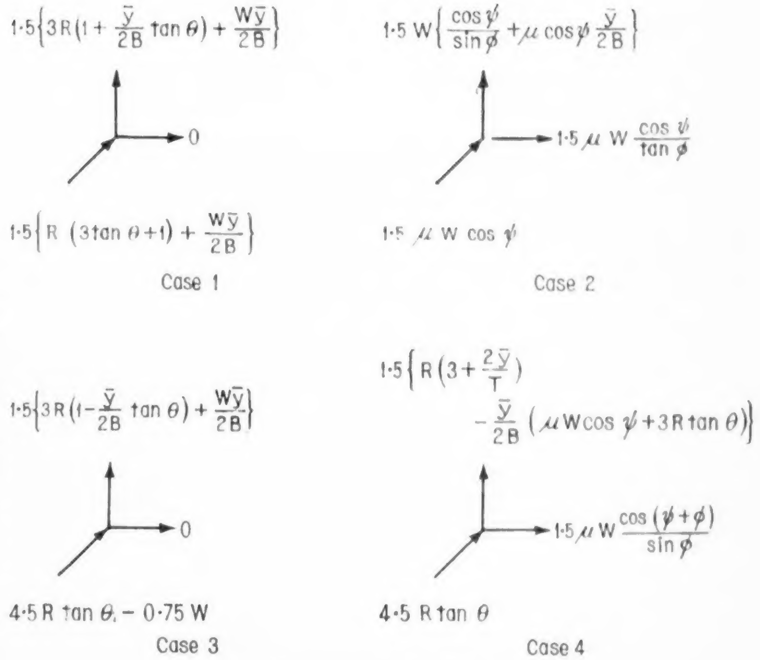


Fig. 5. Load summary for four design cases

The effect of engine torque in the transverse plane may also be included although, in most cases, this is relatively small and may be neglected. Brake torque might introduce critical loads for some structural members. It may, therefore, be necessary to consider the case of hitting a bump whilst braking. This is done simply by reversing the signs of the loads due to acceleration. The result will probably be an increased rearward component and a decreased vertical one.

Case 4. *Hitting a bump whilst cornering*. This will also be a design case for the rear end. The bump loads act along a line parallel to the longitudinal axis of the car. They do not, therefore, have any effect on the side loads as at the front end where, in a turn, the wheels are not parallel to the longitudinal axis.

(a) Vertical load due to bump $= R\times4{\cdot}5$

(b) Rearward load due to bump $= R\times4{\cdot}5\,\dfrac{\sin\theta}{\cos\theta}$

(c) Roll-on load due to bump $= -R\times4{\cdot}5\dfrac{\sin\theta}{\cos\theta}\times\dfrac{\bar{y}}{2B}$

(d) Roll-on load due to centripetal force $= 1{\cdot}5\times2R\dfrac{\bar{y}}{T}$

(e) Side load due to contripetal force $= 1{\cdot}5\,\mu W\,\dfrac{\cos(\psi+\phi)}{\sin\phi}$

(f) Roll-on load due to rearward component at the front wheels $= -1{\cdot}5\,\mu W\cos\psi\,\dfrac{\bar{y}}{2B}$

Then, synthesizing these load components, we have:

Vertical component

$$= R\times4{\cdot}5+1{\cdot}5\times2R\,\frac{\bar{y}}{T}-1{\cdot}5\,\mu W\cos\psi\,\frac{\bar{y}}{2B}-R\times4{\cdot}5\frac{\sin\theta}{\cos\theta}\frac{\bar{y}}{2B}$$

$$= 1{\cdot}5\left\{R\left(3+\frac{2\bar{y}}{T}\right)-\frac{\bar{y}}{2B}\left(\mu W\cos\psi+3R\tan\theta\right)\right\}$$

Rearward component

$$= 4{\cdot}5\,R\tan\theta$$

Lateral component

$$= 1{\cdot}5\,\mu W\,\frac{\cos(\psi+\phi)}{\sin\phi}$$

Again, engine torque can be taken into account if necessary.

These four cases are probably all that are needed to check the structure for strength. However, further experimental work may well show that modifications are needed to the formulae. For instance, it is possible that the roll-on due to the rearward component of the bump load is not very severe owing to its being reacted initially by the inertia of the sprung mass.

The object of this article has been to demonstrate how a system of design loads based not on haphazard guesswork, but on experimental results can be calculated to enable design for strength to be developed into a systematic process giving precise results. The loads given are obtained from data at present available, and it is hoped that in the future more research will be carried out to improve upon the formulae given, and to evaluate more exactly the applied loads. However, the formulae given here have been applied to design a modern car, and have given very good results, reducing weight and economizing in materials.

Strength calculations are not as laborious as at first sight they may appear. A little thought will indicate which design cases to apply, and two cases only are usually all that need to be considered for any one component. Having developed the necessary formulae in the manner indicated, it is only necessary to substitute values for the symbols ; and a diagram, such as Fig 5, can be used to show the magnitude of the loads applied at the wheels. It is then necessary to trace the path of the loads through the suspension components and the structure. The principal advantage of these methods is that they ensure that a load applied through the upper wishbone, for instance, is adequately catered for, not only in the adjacent structure, but also in more remote parts such as the dash and front pillars. Details of how this work can be done will be given in an article to be published later.

The final stage is testing the prototype which, since the loads to be applied are known, is a comparatively simple matter. It may be carried out by distributing sand bags in such a manner and quantity over the vehicle as to give the effect of the vertical loading. Then the side and fore-and-aft loads may be applied with their lines of action passing through the centre of gravity of the vehicle. This can be done by attaching the loading cables to a timber framework fitted across the door pillars. Although this will result in an unduly high concentration of load at the pillars, it will be a fairly reliable test of the structure around the suspension attachment points. Brake torque can be superimposed by means of a loaded arm bolted to the wheels. For this purpose, the wheels must be resting on rollers. Fatigue tests should also be carried out by running the wheels, at the front and rear end in turn, on rotating drums fitted with suitable lobes or bumps. A great deal of research is needed to correlate fatigue test rig results with those obtained in service. This could lead to appreciable time saving in testing.

PLASTICS DIES FOR STEEL PRESSINGS

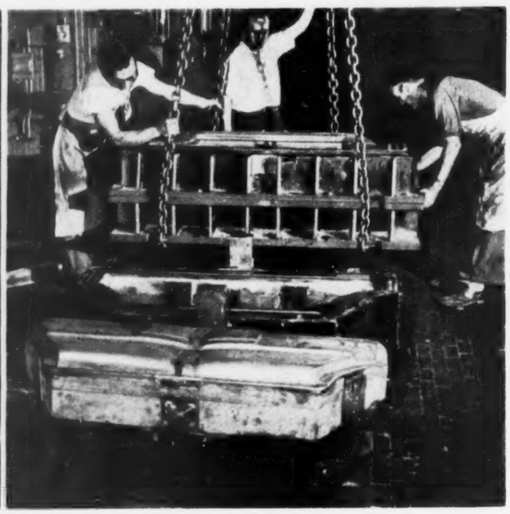
PLASTICS dies have recently been used by the Chrysler Corporation for a factory run of several hundred steel panels for Dodge trucks. A stack of the finished panels is shown in the illustration on the left. Although similar dies have been employed in the aircraft industry, it is claimed that this is the first time plastics have been used for pressing large steel panels for motor vehicles.

There are several economic advantages afforded by this method. Conventional dies of steel weigh about 6,000 lb, whereas a plastics substitute weighs less than 1,500 lb, and takes only three weeks to make. This short manufacturing time was possible mainly because no machining operations are needed with plastics. The economies in both time and labour costs for die preparation are considerable.

First, a plaster mould of the panel was made. Cold thermosetting phenolics plastics was then poured into this to form one half of the two-piece die. Hardening was effected by a chemical action, which generated heat within the plastics. Infra red rays were used in the curing process, which took only 24 hours.

A wax coating, the thickness of the steel panel, was then applied to the mould so that it could be used for casting the second half of the die. This half was hardened and cured in the same way as the first. In order to protect the plastics from fracture or deformation by the high impacts imposed during the operation of the 100 ton press, each part of the die was set in a box made of boiler plate. The halves of the die are shown open in the illustration on the right. In the foreground is the plastics punch. Behind it is the bottom stationary half of the die, and above it is the top half, being lowered to the floor. This development merits serious consideration, particularly for relatively short-run work.

Panels produced from plastics dies



The component parts of the plastics dies

SHELL MOULDING

An Important Foundry Development

THE shell moulding process is one of the most important and valuable developments that have taken place for many years in foundry practice. Johannes Croning of Hamburg introduced the process which was originally known as the "Croning" or "C" process. In this country three important patents in connection with shell moulding were filed early in 1947 by Polygram Casting Company Limited, Power Road, Gunnersbury, London. This company had the full benefit of Herr Croning's exclusive advice, but they eventually developed a distinctive technique known as the Polygram process, which is now more usually designated "shell moulding." Typical castings are shown in Fig. 1.

Fig. 1. Typical shell castings

It is from the shell-like nature of the moulds and cores that the designation derives. The shell is actually built up round the basis of a normal metal pattern, see Fig. 2, or core box. Any of the metals in general use for pattern plates and core boxes can be used for shell moulding plates and boxes. Cast iron, steel and copper-based materials will give plates and boxes that have lives of almost indefinite duration. Aluminium, on the other hand, although it is light to handle, has certain disadvantages as a material for plates and boxes. It is readily etched by the ammonia given off during curing so that the surface is eventually spoiled. Secondly, it is soft and easily abraded by sand or marked by hard tools. It also has a high coefficient of thermal expansion and has a tendency to warp and distort in normal service.

Hand Moulding

Both hand and machine moulding can be employed for the Polygram process. For the simpler form, that is hand moulding, only little equipment is needed in addition to the necessary pattern plates and core boxes. As a preliminary to the production of a mould, the pattern plate is first pre-heated to approximately 160 deg C. The plate is then coated by brushing or spraying with a suitable stripping agent to prevent the mould from adhering to the plate. Following this the pattern plate is placed with its face downwards over the mouth of a dump box. This dump box is essentially a deep bin containing a layer of sand mixed, among other things, with phenolic resins. It is trunnion mounted so that it can be inverted to "dump" the sand mixture over the face of the pattern plate. The operation sequence is shown in Figs. 3, 4, 5, 6, 7 and 8.

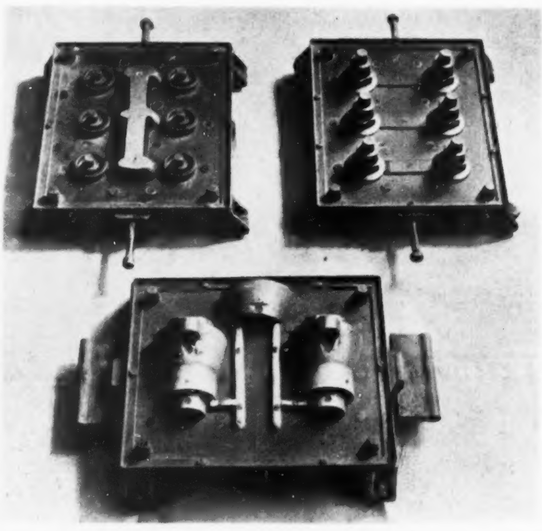


Fig. 2. Pattern plates from which shell moulds are made

Because of the heated condition of the pattern plate, the resin content of the sand melts and an even layer of sand is built up over the surface of the plate. After about 20 seconds the shell will be built up to about ¼ in thickness.. The dump box can then be turned back to its original position to allow the surplus sand mixture to fall back unchanged to its original position in the bottom of the box.

The plate with its half mould is then released from the dump box and transferred to the curing oven. At oven temperature between 350 and 450 deg C, the curing time will generally be in the order of 90 to 120 seconds. This period is long enough to allow the moulder to deal with a second plate while the first plate is in the oven. Output rates are very high. For example, with a 10 in × 12 in mould, the time of 90 seconds for a half mould is enough to allow a fatigue allowance, since it gives the operator a short break in each cycle. An additional personal allowance of 10 per cent is sufficient to establish the standard time. In other words, a single operator can turn out 36 half moulds or 18 complete moulds per hour.

Mechanized shell moulding

For the automobile industry, and for other industries concerned with large quantity production, recent developments in mechanized shell moulding equipment, which have been made by Polygram Casting Company Limited are of great interest. This mechanization not only greatly reduces the man-hours needed for a given output, but also standardizes many of the variables to give a consistent product of even higher quality than is possible by hand moulding.

All the operations employed in the Polygram hand process can be mechanized by means of specially designed machines such as the Polygram automatic shell moulding machine Mark IV, which is illustrated in Figs. 9, 10, 11, 12 and 13. This machine has a manual roll-over of the counter-balanced investment bin.

Polygram shell moulding—operation cycle for manual process

Fig. 3. Applying stripper agent to the pattern plate

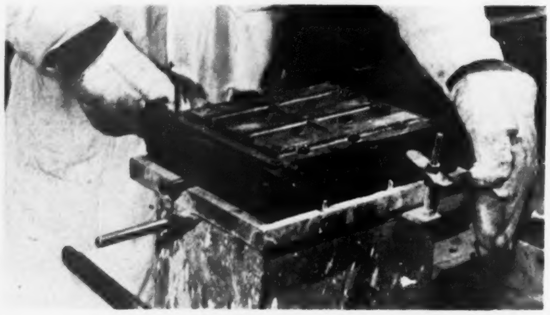


Fig. 4. Inserting the plate in the mouth of the investment bin

Fig. 5. Investing the pattern with the sand-resin mixture

Fig. 6. Pattern plate with the mould ready for curing

Fig. 7. Ejecting the cured half shell

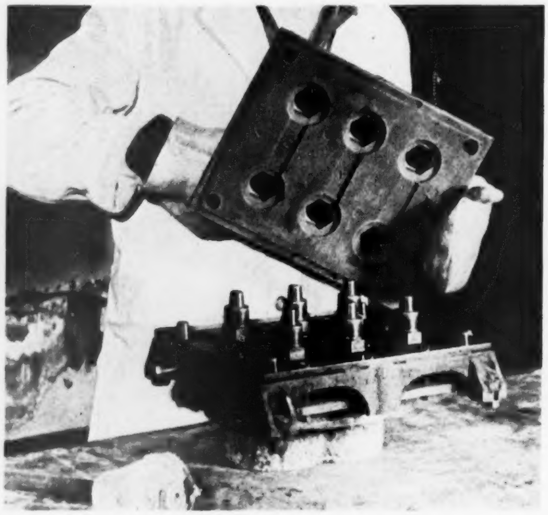


Fig. 8. Half shell ready for assembly

Other designs are available with different bin sizes to suit different pattern plates and a Mark V fully automatic roll-over.

This mechanization removes the actual control of the basic elements from the operator. Instead they are controlled by the instrumentation of the machine. For example, the oven temperature is kept accurately constant by means of a thermostat, while the cycle time is controlled within limits of plus or minus 20 per cent by means of a variable speed handwheel at the base of the machine. In the automatic roll-over machine, the time of investment is, however, still adjustable by the operator, since this gives a certain flexibility that allows four widely differing pattern plates to be used at the same time.

The machine has four stations, at each of which the mould is worked upon. This means that the Polygram automatic shell moulding machine Mark IV has an output equal to that of four ordinary operators. The average time for a complete cycle is 120 seconds. During this period no less than four half moulds are produced, so that the hourly output is 60 complete moulds.

This output has been obtained by reducing the indexing time to one-quarter of the total time, which gives the optimum effective production time for a given cycle. The arrangement is also such that the pattern plate and "biscuit" spends no less than three-quarters of the cycle time in the oven. As a consequence adequate curing time is assured without having to employ excessive oven temperature. For an output of one mould per minute, the time in the oven is 1½ minutes, which is approximately the same as for the hand process.

A feature of the machine that has reduced the cycle time and improved the quality of the biscuit, is the provision of automatic ejection. All the operator has to do is to lift the complete biscuit from its raised position on the ejector pins. The pattern plate is then sprayed with stripper, following which it is invested with the sand-resin mixture ready for another cycle. After investment, the operator swings the plate back into the indexing frame, which carries it into the oven.

Normally, the cycle time is long enough to allow a deft operator to pause for a few seconds between plates. This in great measure eliminates fatigue, and allows an output very close to the nominal one mould per minute to be maintained throughout the whole working day. To obtain maximum output, the oven may be started by a time switch shortly before work commences. It is then only a matter of dropping the required pattern plates into their carriers and running the machine for about ten complete cycles to get it ready for operation. Pattern plates can be changed without slowing the machine, and if they have been pre-heated, the changed plates can be put into operation with the loss of only one half mould for each plate.

The dimensions of the Mark IV machine are 6 ft 6 in overall diameter by 4 ft 3 in to the crown of the oven dome. All main supplies can be brought up under the base of the machine, so that there is free floor space all round. This makes sweeping and maintenance very easy since there are no pipes to harbour dirt. Another factor that makes for a degree of cleanliness not usually associated with a foundry is the complete freedom from any deposition of the sand-resin mixture on to the floor when the bin is inverted.

In the design of this machine, particular attention has been given to accessibility. The specially designed gearbox and the robust and accurate indexing mechanism are designed and constructed to withstand the most arduous foundry conditions. The ejection system was developed after many experiments. These features are such that the only attention the machine should need is a short periodic maintenance check each week and a slightly more comprehensive check at longer intervals. Should a pattern plate be defective, it can be dismounted in a matter of seconds and replaced by a good one without in any way affecting the operation of the machine.

Gas consumption has been kept low by using carefully designed burners, by lagging the oven, and by the filtering of a gas governor. For the first 20 minutes of operation, while the oven is being brought up to curing temperature, gas consumption will be in the order of 200 cubic feet. Subsequently it will be less than 150 cubic feet per hour for the remainder of the working day. The action of the governor is such that the temperature of the oven is rapidly returned to the desired point when the indexing mechanism brings in the cold materials.

A 1 h.p. screen-protected squirrel cage motor drives the machine. It is protected from overload by a safety clutch on the main shaft drive. Should any mechanical defect occur, this clutch will slip, and when the obstruction is removed the machine will again function correctly. A further safety device is incorporated whereby the motor is automatically cut out if the pattern plate is not returned to its carrier before the indexing motion starts.

Shell moulding, even by the hand process, offers many advantages over conventional methods for many applications. Mechanized shell moulding offers even more advantages. It is claimed that the machine will normally pay for itself within a year. It will also ensure the production of consistent moulds, which in turn give consistent castings. Moreover, the machine will set the pace for the rest of the foundry operations so that the problems of production and budgetary control are greatly simplified. Furthermore, the machine can be operated by unskilled labour and the actual work is clean and not too arduous. This will make recruitment of labour for the foundry industry, at present a serious problem, both simpler and cheaper than it has been in the past.

It would appear that these developments in mechanization bring the Polygram shell moulding process into a more than competitive position in relation to conventional sand moulding for many automobile components. It is claimed, for example, that the Mark IV machine will give at least twice as many moulds per hour as a sand moulding machine. Moreover, the capital cost of the machine is, if anything, less than that of conventional equipment. Its use also eliminates all the ancillary equipment

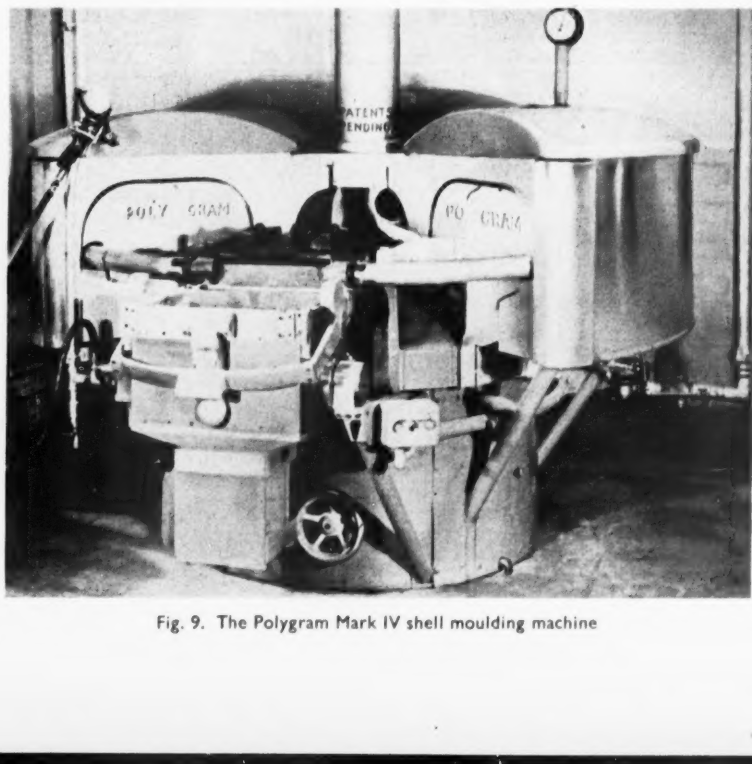


Fig. 9. The Polygram Mark IV shell moulding machine

Polygram mechanized shell moulding

Fig. 10. Pattern plate ready for investment

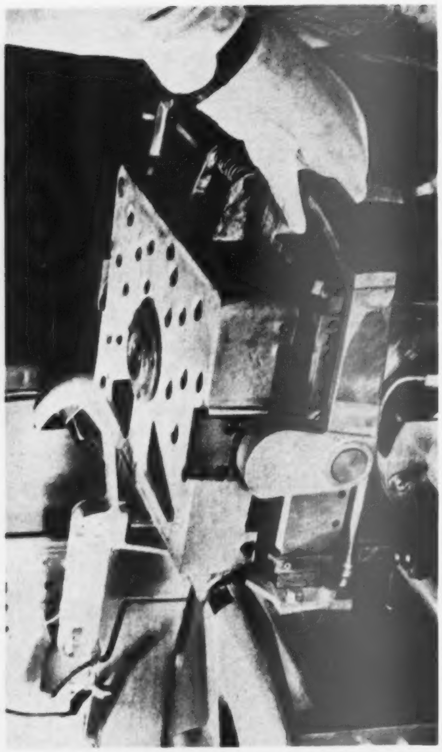


Fig. 11. Plate clamped in mouth of investment bin

Fig. 12. Invested plate in position for the curing cycle

Fig. 13. Cured biscuit ready for ejection and assembly

normally required, such as sand conditioning plant, moulding boxes, large sand handling plant and extensive pouring and knock-out areas. Shell castings virtually knock themselves out since the molten metal ignites the resin binder in the sand during pouring, and the castings merely need to be lifted out of the remains of the mould. The casting is free from fine and adherent sand and requires but little fettling.

For many automobile components dimensional accuracy of the casting is of primary importance, and in this respect shell moulding is remarkably good. For example, limits as close as plus or minus 0·002/0·003 in can be maintained. In addition, the castings have uniform hardness and are free from hard spots, factors of great importance for economic machining. It is also possible to cast thin sections, such as the fins on air-cooled cylinder blocks, without chilling.

Further details of the process can be obtained from Polygram Casting Co., Ltd., Power Road, Gunnersbury, London, W.4. For those who wish to produce shell castings for incorporation in their own products, the use of Polygram patents and complete technical information concerning all aspects of the process, including mechanization, can be obtained by the payment of a small royalty. Others can obtain details of suitable Polygram licensees.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Lighting in Industry

London: BRITISH ELECTRICAL DEVELOPMENT ASSOCIATION, 2, Savoy Hill, London, W.C.2. 1953. 5½ × 8¾. 154 pp. Price 8s. 6d.

Factory lighting can have a very considerable effect on productivity, but it does not always receive the attention that it merits. Some of the questions that must be answered are:—In what degree does lighting affect individual output?—What is the difference between good and bad lighting?—How can existing lighting be improved and yet be economical in operation?—On what data should lighting systems be planned? This book provides the answers to these and many other questions relating to lighting in industry. In addition, it provides the necessary data for lighting design.

There is a valuable section which gives a detailed account of the method of conducting a lighting survey. This will allow managements to survey their own factories to determine whether the lighting is effective and used efficiently to ensure "enough light in the right places at all times."

Automobile Body Reconditioning

By Theo. A. Wohlfeil, Erik E. Frisk and A. B. Saxman.

New York: MCGRAW-HILL BOOK COMPANY, INC. Available from the McGraw-Hill Publishing Company Ltd., 95, Farringdon Street, London, E.C.4. 1952. 7 × 10. 157 pp. Price 31s. 0d.

The business of automobile body repair and refinishing has come into its own in recent years when new cars have been difficult or impossible to obtain. These times may not occur again for a generation but, nevertheless, there must always be those engaged in this work. To execute this work correctly is a craft which requires the trained services also of sheet metal workers, upholsterers, cellulosers, mechanics and welders. Anyone may be able to knock a damaged motor car body back into roughly its original shape but to restore a body, or chassis, to its original alignment and degree of finish requires a deal of experience.

This is an American book, but the principles of working are similar to British. Probably the types of tools illustrated and described as being those required for the work may not be exactly similar in this country, but the same results have to be secured. A chapter is devoted to welding equipment and how to use it, although we think that the repairer would need to know this part of the work before commencing repair work. He would, probably, have advanced beyond the stage suggested in this book before he commenced to read this work. Nevertheless, the special applications of welding to body repairs are valuable information. Body panel aligning and straightening are covered independently and there is a chapter on the use of hydraulic body jacks. A book could, of course, be written on painting equipment and techniques, but much valuable information is provided in the notes included under this heading. The final chapter covers damage estimating.

Much of the value in this book lies in the very many half-tone illustrations of damaged parts and repairs to damaged parts. The book is full of pictures that express the methods of doing this work much more positively than any amount of text. The book's appeal will probably be to a limited number of readers, and mainly to motor car body repairers, but to these it certainly offers value for money.

Teach Yourself Mechanical Draughtsmanship

By S. M. Hood, M.I.I.S., Aff.I.W.M.

London: ENGLISH UNIVERSITIES PRESS LTD., St. Paul's House, Warwick Square, E.C.4. 1952. 4¾ × 7. 183 pp. Price 6s. 0d.

Most technical books published to-day for as little as six shillings contain little important or educational material, unless they are prepared for school children and, consequently, can be produced and sold in large numbers. This one, however, gives the basic principles of mechanical draughtsmanship and will be distinctly valuable to the junior draughtsman. It will also be understood by the workman in the shop who is sufficiently interested in his job to secure a copy. The author states that this book will enable the junior draughtsman to produce drawings that the man in the shop can follow, and this is quite true, but it is thought that the readership might well extend into the shops in certain cases.

The book is divided into eleven lessons commencing with the equipment that the junior draughtsman must possess or have access to. Use of this equipment is portrayed in the lesson on geometrical application and construction. Thereafter, the book deals with the real training of the draughtsman, commencing with a lesson on projection and continuing with pictorial drawing, which includes isometric projection of solids, oblique drawing and oblique projection. Twenty pages are devoted to sections, including pyramids, hollow cylinders and cone faces at all angles.

The correct drawing of screw threads is valuable to all draughtsmen and included here are most of the standard threads. A similarly valuable chapter is that dealing with nuts, bolts, washers, screws, studs and foundation bolts. Many locking devices are illustrated and described. As expected in a book on this subject, there is a lesson on Limits and Tolerances.

Each chapter concludes with a set of exercises in drawing and a good selection of data tables is included at the end of the book. In short, the book can be recommended to junior draughtsmen in any industry.

THE MANUMATIC CONTROL

Automatic Synchronizing of the Throttle and Clutch Controls with the Gear Shift Operation

THE Lockheed Hydraulic Brake Co. Ltd., of Leamington, have recently introduced a semi-automatic transmission system, known as the Manumatic control. This is yet another stage in the search for an automatic or semi-automatic transmission which is less costly and more efficient than the hydro-dynamic drives currently so popular in the United States. The simplification of driving technique is an aim worthy of careful study, for it offers many positive advantages. While anyone, whether aged or youthful, brilliant or dim-witted, can learn to drive a motor car, a considerable amount of mental alertness and skill are occasionally needed to avoid an accident. Moreover, it is not only to reduce the amount of skill required that an automatic system of control is desirable, but also to reduce driver fatigue on long journeys and in congested traffic.

So far as gear changing is concerned, skill is required because of the need to co-ordinate the movements of the clutch pedal, the throttle control, the hand control for the gear shift operation, and sometimes the foot brake. It is a relatively simple matter to provide an automatic mechanism to co-ordinate the gear shift and clutch operations. However, with such a system it is still necessary to operate the throttle and gear shift lever simultaneously. In the early stages of the development and design of the Manumatic system, it was considered that the only satisfactory arrangement would be one in which control could be effected by the movement of only one lever. Accordingly, a mechanism has been evolved which, when the gear shift lever is moved, automatically operates the clutch and throttle in the appropriate sequence.

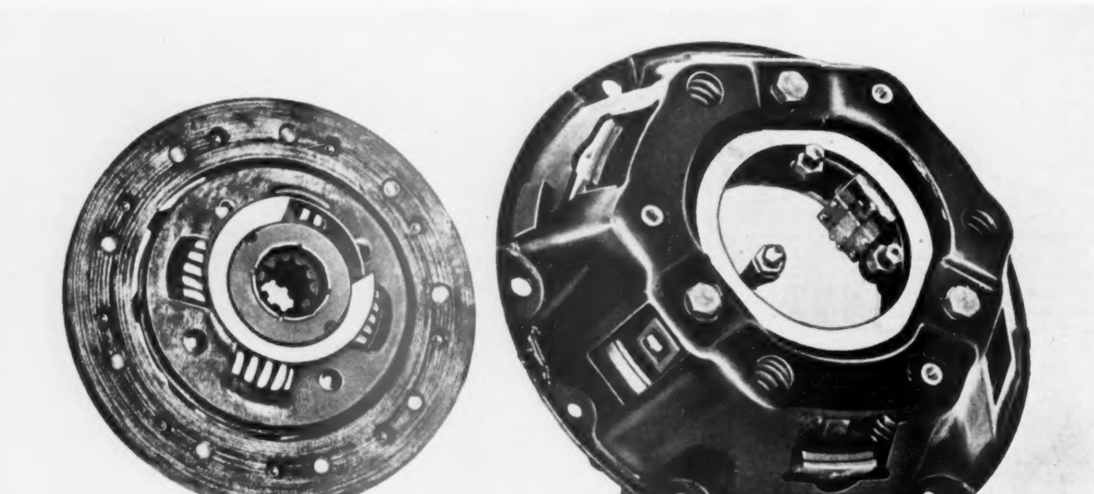
At the commencement of both upward and downward changes, the throttle is closed by a mechanical linkage connected to a clutch-operating servo. After the change has been effected, this linkage also co-ordinates throttle opening operation with the clutch re-engagement. Clutch disengagement and the throttle opening operation for downward changes, are both effected by vacuum servos, controlled by two solenoid-actuated valves housed in one unit. Take-up' from rest is effected by centrifugal clutch-operating mechanism.

So far as mechanical components are concerned, the units already mentioned are the principal features that comprise the Manumatic system. In addition, there are the vacuum reservoir, and the pipe connections through the solenoid valve to the induction manifold and servos. The vacuum reservoir is needed to retain in the system a pressure low enough to operate the servos under all engine running conditions. Correct sequencing of operations is governed by a series of electrical controls serving the solenoid valve.

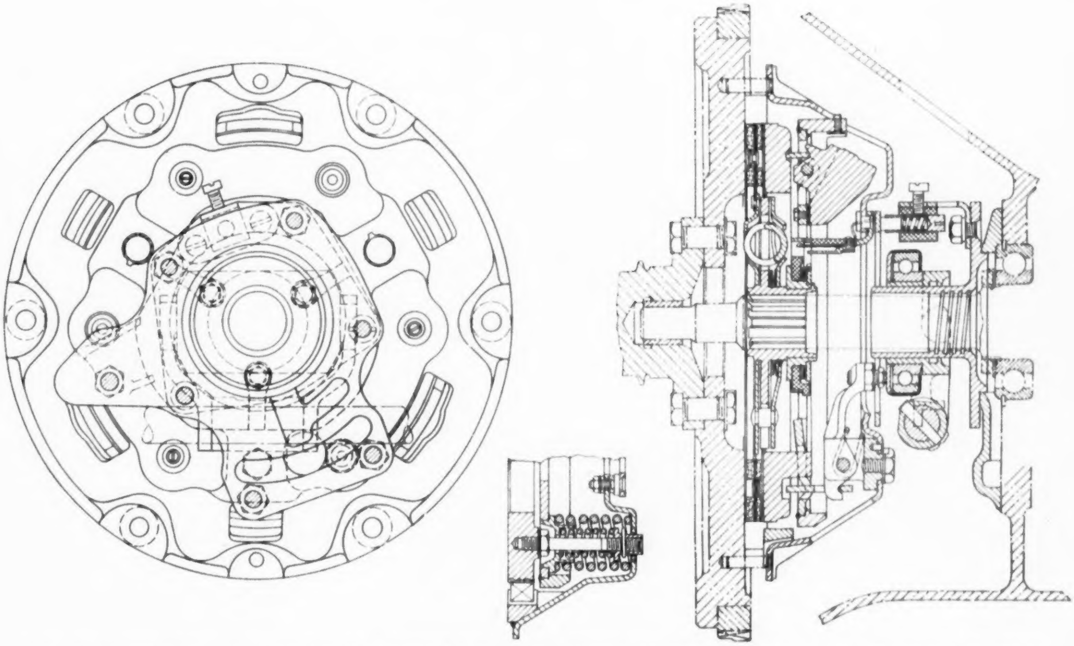
In most respects the clutch is not very different from the conventional single dry plate unit with a sprung centre. The principal additions are a steel thrust plate carrying three bob weights for centrifugal operation, and a synchronizer switch. These components are positioned between the presser plate and the clutch cover, which is bolted to the flywheel.

The clutch and synchronizer switch

The principal component of the synchronizer switch is an eccentric nylon disc carried round the hub of the centre plate of the clutch. It is frictionally loaded against a shoulder on the hub by a crimped spring washer between two plain washers on the back end of the hub. The rearmost washer is peened in position while the front one, bearing on the nylon, has a tab turned back to engage in a slot in the rear one to prevent relative rotation. A brass ring is housed in a groove around the periphery of the disc, and bridges a cut-out in the nylon. Two platinum contacts mounted on the cover project into the cut-out. Therefore, when the clutch centre plate rotates faster than the driving members, it turns the eccentric nylon disc relative to them until the brass ring contacts the platinum points. The electric circuit thus completed energizes the solenoid controlling the throttle servo. This opens the throttle, and the engine speed increases until



The nylon eccentric of the synchronizer switch is carried on the hub of the driven member



In the Manumatic clutch, a thrust plate is interposed between the presser plate and the clutch cover. The scrap view shows the thrust and cushion spring assembly

the clutch unit rotates at a speed slightly faster than the centre plate. Then the eccentric nylon disc and brass ring rotate backwards relative to the clutch casing, the contact is broken and the solenoid valve releases the throttle servo.

Centrifugal loading had to be considered when designing the mountings for the platinum contacts. Each of these contacts is carried on the end of a brass strip about $1\frac{1}{8}$ in long. Both strips are clamped between two blocks of insulating material, from which they project approximately $\frac{1}{4}$ in into the cut-out in the nylon disc. While their outer faces bear directly on the insulating block above them, their inner faces, for a length of $\frac{5}{8}$ in, are about $\frac{3}{32}$ in clear of the lower block. This means that they are flexible so far as the inwardly directed contact forces are concerned, but since they overhang only $\frac{1}{4}$ in from the top block, they have adequate stiffness to resist centrifugal force.

The insulating blocks are carried on a small sheet steel bracket riveted to the clutch cover. One of the contacts is connected to earth on the cover, and the other to an insulated slip ring. Bearing against this ring is a spring-loaded carbon brush assembly on a small bracket bolted to the extension on the gearbox front cover.

Mechanical operation of the clutch is based on the conventional arrangement of a friction-faced, driven plate between the flywheel and a presser plate. Projections cast on the periphery of the presser plate locate it radially in the clutch cover and engage in slots to supply the drive. The cover is bolted and dowelled to the flywheel in the normal manner.

When the clutch unit is only rotating slowly, the presser and thrust plates are held mutually in contact by three light coil spring and tie rod assemblies which are installed coaxially in three of the six main thrust springs. These thrust springs are interposed between the clutch cover and spring seating-washers spigoted into holes in the thrust plate. Each tie rod is passed through a clearance hole in the spring seating-washer, and its front end is screwed into the presser plate. Its rear end is threaded to carry a special nut which projects through a clearance hole in the clutch cover. This nut is provided so that adjustment can be made to the compression in the light spring, which is clamped between it and the seating-washer on the thrust plate. The compression in this spring governs the take-up speed of the clutch. When the adjustment has been made it is locked by a split pin.

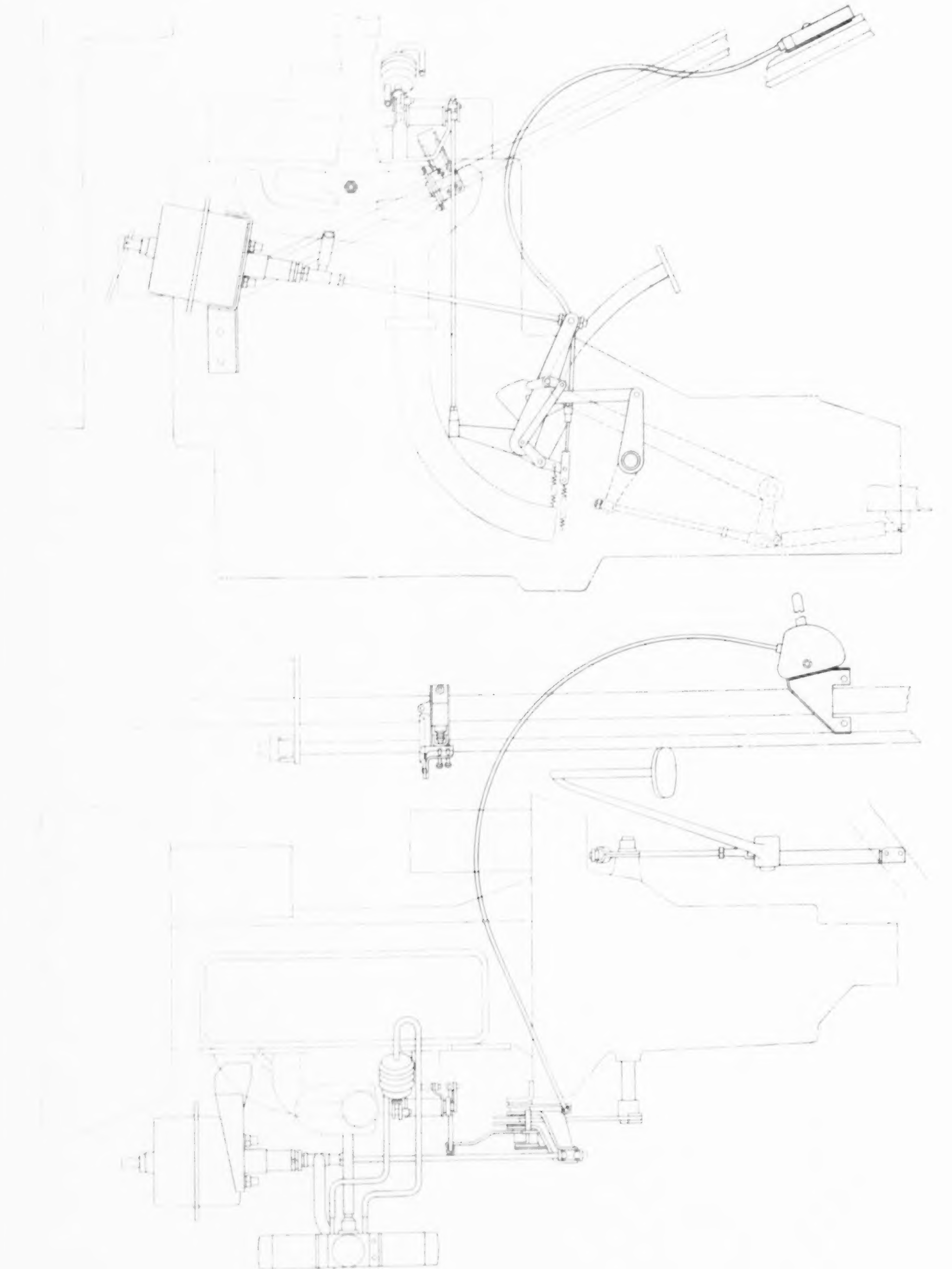
As the engine speed is increased, the bob weights pivot under centrifugal force, and bear on the presser struts to move the presser plate forward, relative to the thrust plate, until it makes contact with the centre plate of the clutch. After all clearance is taken up and the cushion springs are compressed, further outward movement of the bob weights forces the thrust plate to the rear against the action of the main thrust springs. The reaction of these springs is, of course, taken through the weights and presser struts to hold the clutch firmly in engagement. Finally, the weights come up against the thrust plate and no more movement is possible.

The thrust plate may also be moved to the rear by a mechanism similar to that normally employed for clutch disengagement. When operated in this way, it is moved to the rear a distance great enough for the presser plate to be withdrawn clear of the centre plate regardless of whether or not the bob weights are fully extended against the stops. To effect the withdrawal, a thrust bearing in a carrier, round a sleeve extension bolted to the gearbox front cover, is moved forward by the operating fork until it bears against the hardened pads on the inner ends of three clutch withdrawal levers. These levers are pivoted on pins carried in dowel-located lugs which are spigoted and bolted to the clutch cover. Each is grooved at its outer end to seat in a slot in the rear end of a tie-plate. The front end of the tie-plate is T-shaped. On assembly it is passed from the rear through a radial slot in the thrust plate, and then turned through 90 deg so that the arms of the T rest in grooves on each side of the slot. A hairpin spring round the pivot pin of the lever ensures that the tie-plate is at all times loaded in tension, otherwise it would fall off the end of the lever.

The clutch servo

Two main considerations governed the design of the clutch servo, so far as its functioning was concerned. One was that the withdrawal action must be rapid to avoid dragging and wear, as well as to give speedy operation, and the other was that a two-stage return action is desirable. In the first return stage the action must be rapid, again to give a speedy operation, but in the second or final stage, a slower action is needed to avoid jerky engagement.

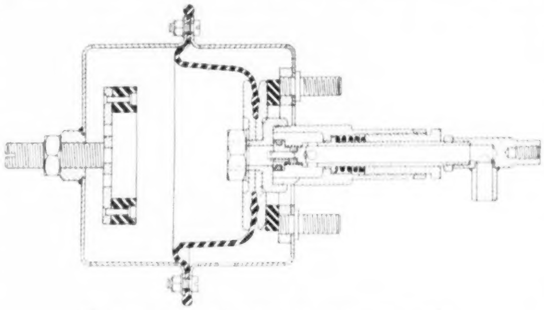
A flexible diaphragm type of servo has been adopted in preference to a piston and cylinder arrangement which



ARRANGEMENT OF THE MANUMATIC CONTROL SYSTEM

would have called for close manufacturing tolerances. The body of the unit consists of two cupped steel pressings secured together, to form a closed cylinder, by 3/16 in diameter bolts passed through flanges around their rims. Clamped between these flanges is the diaphragm, which is also cup-shaped to permit the operating rod a total axial movement of about 2⅛ in. The inside diameter of the cylinder is 4 5/16 in.

In the centre of the diaphragm is a hole, so that it, together with two 3 in diameter discs, may be passed over the threaded end of the operating rod. The discs are mounted one each side of the diaphragm, and the whole assembly is pulled against a shoulder on the rod by a nut on the end. In addition to their function of sealing the centre of the diaphragm, these discs also form the stop, and limit the movement of the rod by bearing on a rectangular section, rubber ring at each extreme limit of its travel.



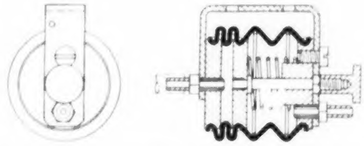
The clutch servo has a two-stage return action

One of these rings is riveted to a circular reinforcing plate round the hole, in the end of the cylinder, through which the rod is passed. This plate is also drilled and tapped to carry the studs by means of which the unit is attached to its support bracket. The other rubber ring is riveted to an adjustable stop plate at the closed front end of the cylinder. The stop plate is peened on to the end of a threaded rod, which has a screwdriver slot in the other end. This rod is screwed into a nut welded to the centre of the end wall of the cylinder; when adjustment has been made it is secured by a locknut. Thus, this adjustment determines the disengaged position and the total length of travel. Adjustment in the engaged position may be made at the threaded rear end of the operating rod. This end is passed through a clearance hole in a transverse pin in the end of the twin operating lever. It is secured by two locknuts seating on saddle pieces, one each side of the pin. The twin lever is fabricated from two ⅛ in thick strips appropriately shaped, and spaced apart by distance pieces so that the rod can pass between them. Rivets are employed to hold the assembly together.

A valve giving the two-stage return action is incorporated in the operating rod. The ¼ in diameter solid portion of the rod is screwed into one arm of a T-piece. The leg of the T is drilled to form an adaptor on which is fitted the end of the flexible hose connecting it with the solenoid valve, and the other arm is drilled out to communicate with the adaptor. This arm is also internally threaded, and is screwed on to the rear end of the ½ in diameter stem of the valve and secured by a locknut. The front end of the stem extends into the valve chamber in the body, and is machined down to ⅛ in to carry the valve head. An axial hole is drilled from the rear through most of the length of the stem. Near the front end, this hole is joined by radial holes connecting it with the valve chamber in the body. Thus, manifold depression may be communicated through the T-piece and stem to the chamber.

During assembly, before the T-piece is screwed on, a tubular guide is slid on to the stem together with a coil spring which is retained between the front end of the guide and a collar round the forward portion of the valve stem. The guide is next screwed into the rear of the body of the valve until the collar on the stem comes against another collar in the body. Further tightening compresses the spring, and when a predetermined setting has been reached, a ring nut round the guide is tightened against the valve body to lock the assembly.

The valve head is circular in shape, and carries a rubber seating washer on a shouldered groove around its periphery. It is free to slide on the extension of the stem on which it is carried, except insofar as it is held in the forward position by a compression spring between it and the shoulder where the extension joins the main part of the stem. A split pin through the extension prevents the head from coming off. When seated against the port in the screwed-on cap which forms the end wall of the chamber, the valve is pushed to the rear a short distance along the stem extension,



The throttle servo has a spring-return action

against the action of the spring, so that it is just clear of the split pin and may be further withdrawn, to open the port, by induction pressure. The screwed-on cap on the front of the body incorporates a tubular threaded extension which carries the diaphragm of the servo, and through which is communicated the depression to operate the unit.

The functioning of the servo is simple. When intake depression is communicated by the solenoid valve through the hollow valve stem of the servo into the chamber in the body, the valve head is lifted off its seat. Thus, the depression is passed into the servo clyinder and moves the diaphragm forward, thereby putting the rod and valve unit in tension. The first part of the movement compresses the spring between the guide and the valve stem. Next the rod, pulling against the action of the clutch springs, disengages the clutch.

To re-engage the clutch, the solenoid valve allows atmospheric pressure to replace induction pressure in the system. The first stage of the return motion of the diaphragm is rapid, since it is effected by the relatively powerful clutch springs operating through the control linkage. However, before re-engagement is complete, the valve of the servo seats on its port in the end cover of the valve chamber. Then, atmospheric pressure can return to the cylinder only through two small apertures, one is the very small clearance between the valve head and the valve stem extension, and the second is a metering hole drilled through the valve head. Thus the final stage of the clutch engagement is effected without snatch.

The control linkage

The arrangement of the system of levers and ties is complex. As has already been stated the piston rod of the clutch servo is carried between the outer ends of the nearly vertical, twin lever which is 3¼ in long. A plain lever, about 1¼ in long, but extending forward, is attached to the outer face of the twin lever to give a bell crank effect. Pinned to the front end of the plain lever is the lower end of a tie rod operating the throttle. The upper end of this rod is passed through an eye in the throttle operating lever adjacent to the carburettor. It is retained by a split pin through the end of the rod above the eye. Thus, when the clutch servo is actuated, the throttle operating lever is pulled down to close the throttle. Then, when the servo is returned to the clutch engaged position, the rod slides up through the eye without opening the throttle, and the throttle control is returned to the driver.

Between the two plates of the twin lever, and on the same pivot pin, is an independent lever, about 1½ in. long, connected by a tie link to the actuating lever on the clutch cross shaft. The independent lever extends slightly forward of the twin lever; a pawl is interposed between the two. Thus, when the twin lever is pulled forward by the clutch servo, it carries with it the independent lever and the clutch actuating lever. Likewise, all three levers are pulled back together by the clutch return springs.

The pawl is centrally pivoted on the twin lever. It is actuated by a tie, the lower end of which is pinned to another lever. This second lever is also centrally pivoted and its other end is attached to

a Bowden cable control. When the pawl is in its normal engaged position, the pin connecting the lower end of the tie to the pawl operating lever is co-axial with the common pivot for the main system of levers. This geometric layout is essential in order that there is no pawl action incidental to the movement of the main levers. However, when the Bowden cable control is pulled, the tie rod moves downwards and the pawl swings upwards to the disengaged position, so that operation of the clutch servo causes only the twin lever to be moved, and leaves the independent lever behind.

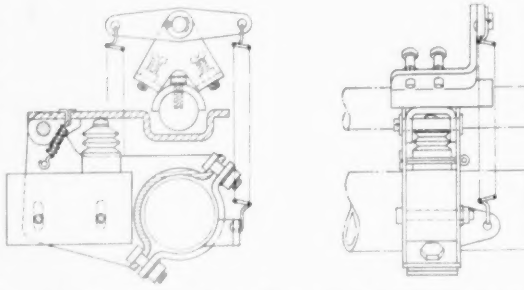
Throttle servo

As in the case of the clutch servo, a bellows type of throttle servo is employed in preference to a piston and cylinder unit because the manufacturing tolerances do not have to be so closely controlled. The function of this unit, which is mounted in a U-bracket on the engine, is to open the throttle for downward gear changes. The bellows unit consists of two circular end plates about $1\frac{1}{2}$ in apart and a similar centre plate that performs the function of a piston. All three plates are $1\frac{7}{16}$ in diameter × $\frac{1}{4}$ in thick and have grooves around their peripheries. The rubber bellows is secured round these grooves by wires, and is further strengthened and stabilized by wire rings inside.

One of the end plates is secured to the inner face of an arm of the U-bracket by an adaptor for the flexible pipe to the solenoid valve. The adaptor is a tube to the outer end of which the flexible pipe is attached; it is threaded at the other end. Midway along its length is a hexagon collar to take a spanner. The adaptor is passed through a clearance hole in the U-bracket, and screwed into a tapped hole through the centre of one of the end plates.

The other end plate is secured to the opposite arm of the U-bracket by a similar adaptor and a set screw passed through the bracket and screwed into the plate. The adaptor and set screw are diametrically opposite one another on a pitch circle diameter of $\frac{11}{16}$ in. At the centre of the inner face of the plate is a boss, $\frac{1}{2}$ in thick, which is drilled $\frac{5}{16}$ in diameter to form a guide for the piston rod. Seated round the boss is a compression spring which, at its other end, bears against the piston. This spring, which is in the low pressure side of the unit, is compressed when the induction pressure is applied to the piston; when the pressure is released, it returns the piston to the disengaged position. A spring return is used in preference to a double-acting piston because it simplifies the solenoid valve.

Both ends of the piston rod are threaded. A hexagon collar is formed adjacent to the inner end so that a spanner may be used to screw it into the centre of the piston. The other end is passed through the guide-boss in the end plate, and a mushroom-headed, adjustable end-fitting is screwed on and secured by a lock nut. When the servo is applied, the mushroom head bears against a bell crank in the engine control system, and opens the throttle. At all other times the return spring holds the head clear of the bell crank.



On the steering column switch, four adjuster screws give individual adjustment for each gear selection

Steering column switch

This switch is mounted on a bracket, clamped on the steering column, just forward of the dash. It is a simple push button micro-switch, operated by the gear shift lever. As this switch initiates the whole sequence of automatic operations of the clutch and throttle servos, its function is that of an electric master control.

The push button is actuated by a lever with its axis horizontal and at right angles to that of the steering column. One end of this lever is pivoted, on the mounting bracket, to one side of the switch. The other end extends over the button and passes between the gear shift control spindle and the steering column. A fabricated 12 s.w.g. rocker secured by two set screws to the gear shift spindle depresses the lever when each gear is selected. The rocker is double ended, one arm being over the outer end of the lever and the other over a point further inboard on the other side of the gear shift control spindle. Screwed in each end are two adjuster screws with lock nuts, one for each gear. A different screw contacts the lever when each gear is selected. Thus, individual adjustment, appropriate to the movement required for the selection of each gear, can be made to the micro-switch operating mechanism.

To maintain the switch positively depressed when the gear has been engaged, a yoke is centrally pivoted on the rocker above its axis of oscillation. Two tension springs, with their lower ends attached to the mounting bracket, are secured one to each end of the yoke. Thus, when a gear is selected and the pivot point is therefore to one side of its top dead centre position, the tension springs, acting through the yoke, retain the rocker firmly on the operating lever.

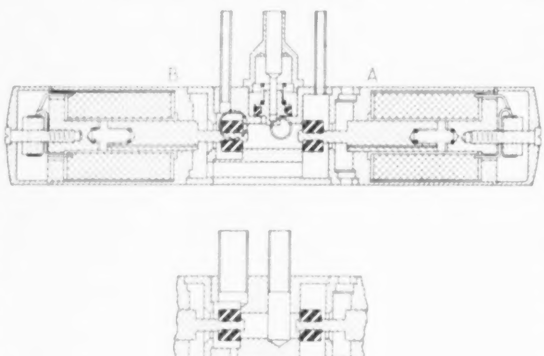
Solenoid valve

In a solenoid valve, as the core piece is withdrawn, the force that can be exerted decreases inversely with the square of the distance. Therefore, for efficient operation, the movement of the valve must be as small as possible. In this unit, it is no more than $\frac{1}{32}$ in.

In general, the unit is built up on a $1\frac{5}{8}$ in × $1\frac{5}{8}$ in × 2 in long rectangular die-casting which, together with two end pieces, forms the valve body. The solenoid windings are housed in cylinders spigot mounted with their axes horizontal, one on each end piece. Both cylinders are $1\frac{5}{8}$ in diameter × $3\frac{1}{2}$ in long, and the inner end of each is flanged to take the four set screws that pass through the end piece into tapped holes in the valve body.

Assembled in the following order into the outer end of each cylinder are: the solenoid winding, an end plate, and an end cap. The end plate is spigoted into the cylinder, and is held in position by four set screws in holes drilled radially through the cylinder walls. A boss at the centre of the inner face of the end plate projects into the tube on which the solenoid is wound. In the end of this boss is an axially drilled hole, about $\frac{1}{8}$ in deep, to carry the end of a compression spring which bears against the valve stem to hold it in the closed position.

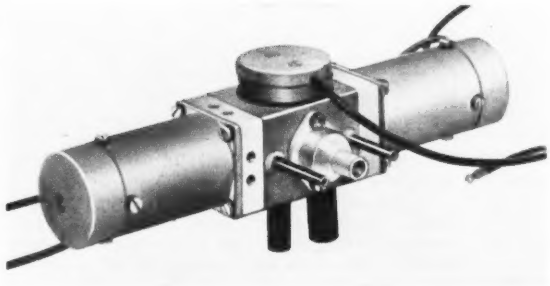
The other face of the end plate is drilled axially at the centre, and tapped for a set screw securing an end cap. This cap protects the electrical connections to the solenoid. The connections are made at a terminal block mounted on the plate, and the internal leads to the solenoid winding are passed through holes in the plate. The



In the solenoid unit, only two valves are employed to open and close four ports

external leads are passed through a hole formed between the plate and a slot in the rim of the cap.

At the inner end, the cylinder is closed except for a $\frac{1}{2}$ in diameter hole through which is passed a $\frac{3}{16}$ in diameter extension of the valve stem. The main portion of the stem is $\frac{1}{2}$ in diameter and forms the core of the solenoid. Around the $\frac{3}{16}$ in diameter portion is a groove, $\frac{1}{32}$ in deep × $\frac{11}{32}$ in long, to carry the rubber valve head. A longitudinal drilling through the $\frac{1}{2}$ in diameter portion of the stem provides an escape for air that would otherwise be trapped between the valve stem and the end plate. The compression spring already mentioned seats in an axial drilling, $\frac{1}{4}$ in diameter × $\frac{11}{32}$ in deep, in the outer end of the stem.



A pressure switch is mounted on top of the solenoid valve

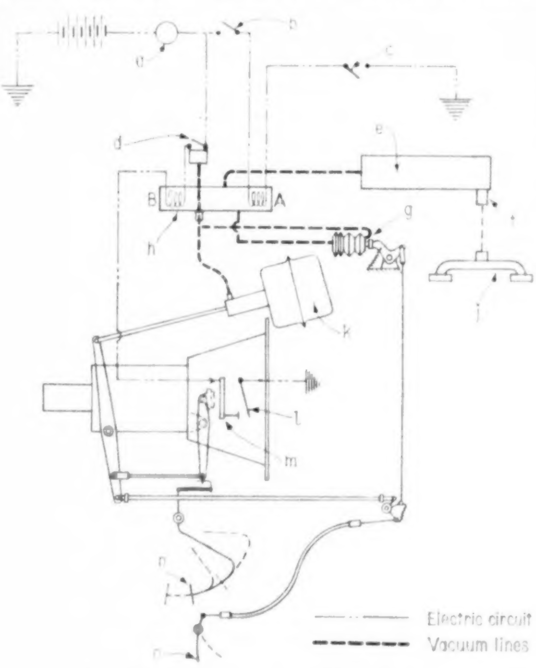
Both end pieces are rectangular in shape, and are spigoted into the centre piece of the valve body. They are $\frac{11}{32}$ in long, and their cross sectional dimensions are the same as those of the centre piece. Each has two functions, namely to form an outer chamber between the cylinder and the cupped outer face of the end piece, and to close the outer end of an inner chamber in the centre piece. From the illustration, it can be seen that the end piece at A is drilled radially so that the chamber between it and the cylinder is open to atmospheric pressure. A short, wire gauze sleeve around the unit covers these radial ports and forms a filter. However, the other end piece, at B, is not so drilled, but has in its vertical face a port communicating with the chamber between the end piece at A and the centre piece.

On assembly, each valve stem is passed through a $\frac{13}{32}$ in diameter hole in the centre of the appropriate end piece, and the rubber head is then put on. Thus, when the solenoid actuates the valve at A, the head seats on the inner face of the end piece and blanks off the hole which is, in effect, a port to atmosphere. At all other times, the compression spring forces the valve away from this aperture and on to a seat in the centre piece to close the induction-pressure port. The action at B is similar except that when the solenoid closes the outer port, it prevents the transfer of pressure from the inner chamber at A to the inner chamber at B.

The centre piece, in addition to being the base on which the unit is built, incorporates all the pipe connections. A $\frac{13}{32}$ in diameter longitudinal drilling forms the centre, or induction pressure chamber which, when the solenoids are not in operation, is closed at each end by the valve heads. A vertical drilling, $\frac{1}{8}$ in diameter, communicates with the longitudinal hole, and carries a short length of sweated-in copper tube forming the adaptor for a flexible pipe to the induction-pressure reservoir. Another tube, $\frac{1}{2}$ in outside diameter, is mounted in a similar manner, alongside the pipe to the reservoir. It gives communication between the inner chamber at B and the clutch servo.

In the side of the induction-pressure chamber, another port communicates with a cylindrical chamber that houses a non-return valve. The axis of this chamber is horizontal and at right angles to that of the centre chamber. The outer end of the non-return valve housing is closed by a screwed-in adaptor for the pipe line to the induction manifold. The non-return valve, which has a rubber head, is held down on the port to the centre chamber by a compression spring between it and the adaptor. A blind, axial drilling is provided in the outer end of the valve to take a guide pin mounted in the adaptor.

Positioned one on each side of the non-return valve chamber are two $\frac{3}{16}$ in outside diameter, short copper tubes sweated into holes drilled through the side of the centre piece into the two inner chambers. These tubes carry the flexible pipe lines to the throttle servo. A depression-operated switch unit is mounted on the side opposite the connections to the clutch servo and reservoir. Induction pipe pressure is exerted on the switch unit through a $\frac{1}{8}$ in diameter hole from the inner chamber at B.



a Ignition switch
b Steering column switch (closed in neutral)
c Hand brake switch (closed with hand brake off)
d Vacuum switch
e Vacuum reservoir
f Non-return valve
g Throttle operating servo
h Solenoid valve
j Induction manifold
k Clutch operating servo
l Synchronizer switch
m Slip ring
n Clutch pedal
o Throttle pedal

A diagrammatic illustration of the control system

Operation of the system

The solenoid valve is virtually the master pneumatic unit for the whole system. Therefore, in order to understand the operation of the system it is necessary to study the illustration of the solenoid valve as well as the diagram of the electric circuits.

When starting from rest, with the hand brake off, the sequence is as follows. The ignition is switched on and connects the battery into the automatic control circuit. When the gear shift lever is in neutral the steering column switch is closed so that the solenoid A is energized. This unseats the valve from the end of the induction pressure chamber and seats it on the port to the outer chamber, which is open to atmosphere. Thus the atmosphere connections to the throttle and the clutch servos are closed and induction pressure is substituted in the system.

The clutch servo moves the clutch control to the disengaged position, and an interconnecting link in the control system closes the throttle regardless of the position of the pedal. Under these conditions, depression of the throttle pedal simply applies a tension to a spring in the control linkage. When the override device is not in operation, the spring is stiff enough to act in the same manner as a rigid link and transmit the control motion to the throttle butterfly.

Induction pressure is communicated to both sides of the throttle servo, that is to say, directly from the pipe at A and also through the transfer passage to the side B, and through the outer and

inner chambers and the pipe at B. Thus, no movement of the throttle servo results. The reason for this seemingly inessential transfer of pressures to the throttle servo will become apparent when the change down sequence is explained. It arises from the fact that only one low pressure connection from A operates the servo during the change down, and to prevent the servo from operating during the upward gear change, the pressures on each side of the piston must be balanced. If one side were opened directly to atmosphere instead of through the balance pipe to the solenoid valve at B, the solenoid valve would have to be more complex.

The next operation in moving away from rest is the shifting of the gear lever to engage first or second gear in the normal manner. The last part of the movement of the lever opens the steering column switch, and the valve at A is released. As a result, the port to the low pressure chamber is closed and the port to the outer chamber, and hence to atmosphere, is opened. The clutch springs then return the unit to the engaged position and the throttle control is released so that the driver has direct control. However, no drive is transmitted, because the engine is running too slowly for the centrifugal mechanism to be effective. Only when the engine speed is increased does the centrifugal clutch engage to take up the drive. This is a useful feature when driving in congested traffic.

Further upward changes may be effected simply by moving the gear shift lever, and the same sequence of automatic operations follows. The initial movement, of course, closes the steering column switch and energizes the solenoid valve to bring into operation the servo which disengages the clutch. At the same time, the interconnecting linkage closes the throttle. Then the gear is selected by further movement of the shift lever, which operates the normal synchromesh mechanism in the gearbox. No skill whatever is required, and the change up is effected as smoothly and swiftly as by skilled operation of a normal gear shift system.

Downward gear changes are effected in a similar manner, but another unit, the synchronizer switch in the clutch, is brough into operation to actuate the throttle servo. The first motion of the gear shift lever into neutral closes the steering column switch. This energizes the solenoid valve A, which causes the servo to operate and disengage the clutch. At the same time the throttle is closed. Then the driver moves the gear lever into the lower gear to effect a normal synchromesh engagement.

The closing of the throttle reduces the speed of the clutch driving member relative to the gearbox primary shaft and clutch driven member. As a result, the contacts of the synchronizer switch lag in relation to the eccentric cam ring which thus completes the circuit across the two points. The solenoid at A is already in operation, so that induction pressure in the inner chambers has closed the depression operated switch and the whole circuit is completed to energize the solenoid at B. There is, however, no immediate change in the position of the servos, since all the chambers to the left of the outer one at A were already subject to induction pressure.

Energizing the servo at B is thus only a preparatory part of the throttle opening sequence. The final movement of the gear shift lever releases the solenoid at A. This admits atmospheric pressure through the pipe to one side of the throttle servo, which then opens the throttle. When the engine and clutch driving-member have accelerated to a speed just faster than that of the driving member, the synchronizer switch is opened and the solenoid valve at B returns to its normal position. This admits atmospheric pressure to the whole system; the clutch engages, and control of the throttle is returned to the driver. The change down is effected smoothly, but it is possible that a very skilled and experienced driver could make a quicker change by double declutching with a normal gear control.

The reason for the incorporation of the depression-operated switch in the synchronizer circuit is now apparent. When running in gear at very low speeds, the centrifugal mechanism might disengage the clutch. Then, if the engine speed were allowed to fall still further, the synchronizer switch would come into operation and, without an open depression-operated switch in the circuit, the engine speed would be increased by the throttle servo and hunting would occur.

Special features

So far, only normal driving operations have been described. However, there are occasions when special conditions give rise to additional requirements. For instance, when the engine is started from cold, with the gear lever in neutral, the driver must have complete control over the throttle. This he would not have under normal running conditions since the steering column switch would be closed, and the clutch servo with its interconnecting linkage to the throttle would be in operation. To provide for cold starting, the solenoid circuit from A is taken through a switch which is open when the hand brake is on, so that the clutch servo and its interconnecting linkage to the throttle are released. This does not affect the starting from rest condition, since the centrifugal clutch is not engaged until the engine speed is increased. Thus, when starting on a hill the first gear may be engaged with the hand brake on. It should be mentioned here that when a member of the staff of *Automobile Engineer* tried a car fitted with this system, a normal start was made without any difficulty on several steep hills.

Nevertheless, on an extremely steep gradient, with the car overloaded, or with the wheels bogged, trouble could be experienced. This is because on most vehicles, the centrifugal clutch would engage at an engine speed below that at which maximum torque is developed. For this reason, the conventional clutch pedal has been retained and this allows the driver to hold the clutch in the disengaged position until the engine speed at which maximum torque is developed is reached.

There are other occasions when direct control over the clutch is needed. One of these is during a tow start, and another is for "inching" in confined parking spaces. When a centrifugal clutch is used, the engine speed has to be increased enough to engage the clutch before the vehicle can be moved. Then, to stop again and disengage the clutch, the brakes must be used to overcome the rotational inertia of the engine and transmission in addition to the translational inertia of the vehicle. This is by no means impossible, but the override operation is, perhaps, more convenient. A third condition when direct engagement of the clutch is required, is when parking in gear instead of using the hand brake. This practice is resorted to in very cold climates in which a hand brake may freeze if left on for long, and it may then be a very difficult and tedious task to free it. For push starting or parking in gear, the clutch can be engaged by operating a lever under the dash facia. The motion of this lever is transmitted through a Bowden cable to the pawl actuating lever.

Nash Developments

MANY mechanical improvements have been made in the 1953 Nash Ambassador and Statesman models. One of the principal advances is the new Statesman Powerflyte engine, which now gives a maximum output of 100 h.p. against the previous figure of 88. The new engine is also said to give greater fuel economy at certain speed ranges. This is a 6-cylinder L-head engine, and the increased horse-power has been gained without any increase in displacement.

The compression ratio has been raised from 7·0:1 to 7·45:1. Another change is that a new double-barrel duoflo carburettor is used. The intake manifold passages have been enlarged and the combustion chambers have been redesigned, and there is a marked advance in engine breathing characteristics through these changes and through improvements in the exhaust system. A new high-lift camshaft is also incorporated in the design.

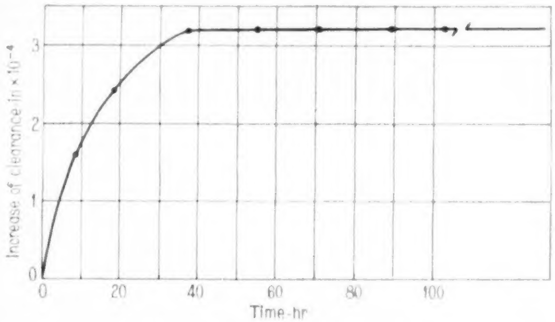
The standard power unit for the Ambassador model is the 6-cylinder 120 h.p. Super Jetfire engine. As an alternative, the Le Mans Dual Jetfire engine may be fitted. This is also a 6-cylinder overhead valve engine. It develops 140 h.p. at 4,000 r.p.m. Two Side-draught carburettors are fitted to work in conjunction with an intake manifold that is larger than that of the Super Jetfire. Power assisted steering is optional for the Ambassador. It gives direct hydraulic actuation of the steering linkage.

WEAR PHENOMENA*

Effects of Lubrication and the Nature of the Superficial Layer

By F. T. Barwell, Ph.D., D.I.C., B.Sc.(Eng.), Wh.Sch., M.I.Mech.E., A.M.I.E.E.

IT is well known that the continued running of a bearing often leads to changes in its performance, which may improve or deteriorate even when external conditions are constant. A typical example of this behaviour is shown in Fig. 1, from which it can be seen that an initially high rate of wear of a bearing gradually decreased, and then sensibly ceased when stable conditions were attained. In contrast to this running-in effect followed by a period of stable operation, there may be mentioned the familiar experience of machine parts that, after having given satisfactory service for prolonged periods, finally wear out and need to be replaced. These three phases of operation: running-in, normal service, and wearing-out, are characteristic of many different types of bearing. Examination shows that these effects are due either to gross dimensional changes, or to changes in the nature of the lubricant or of the rubbing surfaces during running.



Running-in curve for a bearing

Continuous wear

Even the most correctly designed and operated bearing slowly but continuously seems to suffer some wear during operation. This is not necessarily disadvantageous; indeed, running-in consists essentially of this type of wear, and it may often be preferable to encourage this slow wear rather than more severe forms of failure. The main feature of the running-in process is that material is removed in small particles, which are carried away in the lubricant without resulting in any gross surface damage.

The following effects may contribute to the removal of particles:

1. The mechanical interlocking of asperities present on the rubbing surfaces.
2. The localized adhesion of mating surfaces in a manner already exhaustively discussed by Bowden and Tabor[1].
3. The dislodgement from the surface of wholly or partly oxidized particles by the corrosive action of the lubricant or by erosion caused by cavitation.
4. The abrasion of the surfaces by hard oxide particles or by adventitious matter.

It is important to note that dimensional changes in bearings are not necessarily the consequence of wear. The operating normal and tangential stresses may produce plastic deformations in the softer of the two surfaces.

The types of wear discussed above do not lead immediately to failure of the mechanism, and may even continue for many years before their effects are appreciable. Certain forms of surface damage, however, such as scuffing, lead to gross disruption of the mating surfaces. These disruptions may be severe enough to render the bearing surfaces unfit for further use. Typical features of scuffed areas are numerous torn patches and blobs of transferred material. High temperature produced at parts in intimate contact is the cause of scuffing. Under certain conditions of heavy load and high speed, the frictional heat cannot be dissipated rapidly in the surrounding media, and an unstable system is initiated, which leads quickly to localized welding of the opposing surfaces and their subsequent rupture. This type of failure is especially susceptible to prevention by surface treatment applied either before running, or continuously by the addition of suitable chemical compounds to the lubricant.

A common type of failure of rolling elements is characterized by the formation of small pits. The pitting is believed to be due to fatigue of the material in the regions of maximum shear stress which occur at some depth below the surface. This type of failure is sometimes found in ball races.

Abrasion and fretting corrosion

Recent workers, for example Oberle[2] and Brunt[3], associate the abrasion resistance of a material with the amount of deformation it may endure without permanent change. A high value of yield stress combined with a high extensibility are desirable characteristics since they allow the energy of successive minute impacts to be absorbed without damage. If this point of view can be substantiated, the criterion of abrasion resistance should be proof resilience $f^2/2E$. On the other hand, Grodzinski[4] has demonstrated that the resistance of diamond to abrasion depends on the direction of the rubbing action, which indicates that atomic structure also has an effect.

Wear and damage to machine components are usually associated with a comparatively large relative motion of the mating surfaces.

There is, however, a particularly malignant form of damage which can occur when two mating surfaces, nominally at rest with respect to one another, are subject to slight vibrational slip. This phenomenon, which has been termed "fretting corrosion," is characterized by pitting of the surfaces, and the generation of considerable quantities of finely divided oxidized debris.

Owing to the restricted movements of the two surfaces, the oxidized debris cannot easily escape, and extremely localized abrasion results. The formation of the oxide debris does not appear to be dependent upon the generation of high surface temperatures, but rather on the considerable chemical activity of the abraded virgin material, combined with the high pressures at the point of contact. Minute amounts of water vapour between the surfaces appear to play an important role, and in the particular case of steel the characteristic red oxide, known to engineers as "cocoa," is probably formed via the hydroxide phase. Effective oxygen concentration at the surface also plays an important part in fretting corrosion. The protective action of lubricants for instance is enhanced by a low solubility and diffusivity of oxygen in them.

The role of metal oxides

Under very light conditions of loading, the oxide itself may offer a certain amount of protection, and sliding will probably take place at an oxide/oxide interface. Under these conditions, the prevention of metallic welding, however, is dependent upon the relative rigidities of the oxide and the underlying metallic substrate. Therefore it will be affected by the metallurgical changes occurring in the intermediate surface zones of the bearing material.

Recent work by Moore and Tegart[5] has shown that under conditions of repeated sliding, without a lubricant, the continuous deformation of the surface layers can lead to the inclusion of oxide particles to a considerable depth within the disrupted surface zone. Work in the Mechanical Engineering Research Laboratory at Thorntonhall

* *From a paper presented at a symposium, on Properties of Metallic Surfaces, organized by the Institute of Metals, November, 1952.*

has shown this to be dependent to some extent on the velocity of sliding. The effect of a high velocity of sliding is to increase the degree of oxide inclusion. Moreover, the presence of such embedded particles, inhibits any possible future grain growth and effectively stabilizes the disrupted surface layers. We may conclude therefore that under practical conditions of sliding, the bearing surfaces do not generally consist of a simple work-hardened or amorphous structure, but rather of a complicated mixture of strained metallic material, oxides, and other possible decomposition and reaction products.

In the presence of a lubricant, the behaviour of an oxide film and its rate of re-formation will be greatly modified. This is due in part to the reduction in the effective oxidizing conditions, and in part to the additional chemical, physical and mechanical interactions. Experiments similar to those mentioned in the previous paragraph, but with an oleic acid blend as lubricant, showed that the oxide particles were not embedded to the same extent as they were under dry conditions. It appears that the presence of the lubricant reduces re-oxidation and also lessens the deformation of the surface layers. This explanation, however, does not fully account for the behaviour of the oxide particles in continuous sliding and in the presence of a surface-active lubricant, such as oleic acid.

Davies[6] found that, with a refined mineral oil as lubricant, there was evidence that wear was high until a quantity of wear debris, sufficient to fill in the irregularities of the surface, had been accumulated. The wear debris appeared to consist of aggregates of finely divided oxides approaching 1μ in diameter. When a surface-active agent such as oleic acid was added, it effectively dispersed the finely divided oxide, which was prevented from accumulating on the rubbing surfaces. As a result, there was a subsequent increase in the rate of wear. Davies concluded that under certain conditions the chemical reactivity of the oleic acid was of secondary importance compared with its dispersing and detergent properties. Thus it appears that oxidation of the surface not only reduces the possibility of metallic contact, but that when such oxides and other wear debris are removed from the surface, they can assume the additional protective role of filling up the surface irregularities, and thereby reducing the effective load at the asperities.

Gwathmey, Leidheiser and Smith[7-9] have studied the friction and oxidation of single crystals of copper, and have shown that the chemical properties and the processes of friction and wear depend on the crystal face exposed at the surface. When a single crystal was corroded by hot oils, it was found that some faces became very rough, whereas others remained smooth. Friction between bare copper crystals from which all oxide had been removed was bound to vary by a factor of four, depending on the crystal faces involved.

The functions of the oxide in continuous and prolonged sliding can therefore be summarized as follows:

1. Oxide formation in the presence of water may lead to highly reactive free radicals.
2. Under mild conditions of sliding the oxide may prevent metal-to-metal contact.
3. Under more severe conditions and in the absence of a surface-active lubricant the oxide will embed itself into the surface of the metal.
4. In the presence of a surface-active lubricant oxide will detach itself from the surface and promote a mild form of wear.
5. In a polycrystalline material oxidation may lead to roughening owing to different rates of action on different crystallographic planes.

Modification of the bearing surface

Besides the changes that occur in the structure of the oxide layer, the metal itself is also liable to be profoundly affected by rubbing. Finch[10] has surveyed the various factors involved, and so only a few of the more important will be considered here. The main changes are in surface topography and in metallurgical structure. Chemical effects will be discussed under the heading of "Artificially modified layers."

The importance of "running-in" new bearing surfaces by using light loads and copious lubricant is fully recognized by engineers, but the causes of improvement are not generally appreciated. To a large extent they are purely geometric. Thus, the performance, as demonstrated in the illustration, of the journal bearing can be ascribed to slow wear and flow of the soft bearing material to give a geometrical configuration, which promotes the formation of hydrodynamic lubrication. Once this is established, the rate of wear becomes negligible. If the accuracy of manufacture had been sufficient to give hydrodynamic lubrication initially, no wear at all would have occurred. Similarly, though on a microscopic scale, the peaks of the initial irregularities of a plane ground steel surface were smoothed over by rubbing against white metal; once again the resulting geometry of the system encourages hydrodynamic lubrication and a very low rate of subsequent wear.

In contrast to these changes in profile that lead to improved performance, an unstable system that usually leads to complete failure may be considered. Under certain critical conditions, the temperatures at the scattered points of intimate contact are sufficiently high to cause melting. If the volume of metal affected is such that, on subsequent quenching, some of it solidifies as a blob that increases the surface irregularity, then on repeated traverses over the rubbed area the lubrication and temperature conditions may be worsened, more metal affected, and the surfaces rapidly roughened and torn. This is the type of failure designated as scuffing. From a taper section through such a scuffed area on a steel specimen the history of the deterioration may be traced. The initial pick-up between the surfaces is often covered with a further layer at each traverse and gradually forced deep into the bulk of the material. Different etchants and micro-hardness determinations may reveal the very considerable variations in structure throughout the scuff. In one such specimen, the initial hardness of the rider was 200 D.P.N. (30 kg) and that of the slider 150, whereas the hardness of the resolved areas of the transferred material ranged from 350 to 850, indicating very rapid thermal quenching. The presence of unresolved inclusions when a plain mineral oil is used may be indicative of carburization of the steel by decomposition of the lubricant.

Except under fully hydrodynamic conditions of lubrication some inter-oxide and intermetallic contact may always be expected. Under milder conditions than those mentioned in the previous paragraph the heat generated is rapidly dissipated in the surrounding media. However, the high stresses and temperatures at the points of intimate contact may still be sufficient to cause plastic flow of the material. They often give rise to smoothing of the irregularities, polishing of the surface, and distortion of the crystals beneath. The effect on the resultant structure may be profound, especially if oxide particles are embedded in the worked material. As previously mentioned, Moore and Tegart found that grain growth is effectively inhibited by the presence of occluded oxide particles.

The structure of the extreme surface layer, just below the oxide covering, is of great importance in determining the performance of the rubbing surface; its influence may be as great as, if not greater than, the purely geometric effects already noted. Finch has shown how even the lightest mechanical polishing of two anodically polished single crystals of copper can substantially reduce the damage caused by sliding. The presence of this almost amorphous material, or Beilby layer, probably serves to reduce the tendency towards the occurrence of cold welding, and provides stronger mechanical support to the oxide film.

On the other hand, working and strain-hardening of the crystalline material below the surface is liable to increase the susceptibility to frictional damage, for the metallic junctions, once formed, will themselves be stressed and finally broken. Unless the junction is weak owing to a lack in affinity, cleavage will probably occur below the surface, at a point determined by the relation of the modified Hertzian stress distribution to the local strength of the material. The size of the particle removed may therefore depend upon the depth and degree of local variations of surface hardness.

Changes below the surface

If we assume a simplified model of two approaching spherical surfaces to be an approximation to the points of real contact, the stress distribution will conform to the classic analysis of Hertz, in the absence of tangential resistance at the interface. Then, the maximum shearing stress will occur at a point below the interface, and the first yielding or flow is to be expected in this region. With tangential resistance, however, the maximum shearing stresses will occur near to, or even at the interface, and in this case the surface layers will yield first. A graphic illustration of this difference has been given by Moore[11], who found that in unlubricated sliding there was considerable plastic deformation around the track, whereas in lubricated sliding the sub-surface material was deformed although the initial irregularities were relatively unaffected.

Very often the surface stresses are insufficient to cause plastic flow of material except in the extreme surface layers. In repeated cycles, however, lower values of stress may be sufficient ultimately to cause fatigue and resultant failure. This is the basic cause of pitting of rolling bearings and of gear teeth. In order to study this mode of failure recourse was made to a simulation method of test. The apparatus was a Boerlage[12] four-ball machine, modified to allow the lower three balls to rotate freely in a special race. As a result of this modification, the upper ball, on rotation, is subjected to continuous rolling contact with the three lower balls along a circular track. During prolonged running under load, pits develop in the track on the upper ball, and their nature is very similar to those encountered in practice.

Numerous tests were made, using a straight mineral oil as lubricant, with a variety of loads and running periods. For the purpose of this review only the phenomenological aspects of the results will be considered. These were obtained by examining the upper balls from a number of tests, after various running times under uniform conditions. The balls were examined normally, and also in section.

On the balls, the running tracks were clearly visible as stained bands, and after prolonged running the pits formed. Closer examination showed that a certain amount of plastic flow had occurred within the track, which was left as a shallow depression on the ball. At the inside edge of the track, small transverse cracks appeared after a short running period, and these gradually developed as running proceeded until a flake of metal became detached.

Microsections across the tracks showed that associated changes were taking place beneath the surface. The change in microstructure of the steel increases with time. The precise metallurgical structure induced has not yet been positively identified, but appears to be a form of mechanical troostite.

In the later stages of running scattered cracks were observed to form below the centre of the track, in the areas metallurgically altered. These cracks gradually spread in directions parallel to the surface. Some of the failures may also have been a result of the meeting of the surface and sub-surface cracks. Once the surface had started to break up in this way the damage would have quickly spread until the whole running surface became disintegrated.

The loads and stresses employed in these experiments were rather higher than those usually employed in ball bearings. However, that similar results may be expected under conditions more closely akin to practice was shown by preliminary experiments at lower stresses. The cure would seem to be to reduce the fatigue susceptibility of the system, either by attention to the purity and homogeneity of the material, or by closer control over the stress distribution.

Although these fatigue phenomena have been investigated in relation to rolling contacts, it is probable that they also contribute to the slow wear of bearing surfaces. In cases where the conditions of contact are such that shallow sub-surface stresses are induced, but where the values of these stresses are too low to cause plastic flow, it may be expected that prolonged operation will cause fatigue of the surface layers, and their ultimate breakdown in small particles. In this connection it is interesting to note that Rose[13] has experienced slight pitting of steel surfaces after prolonged sliding at relatively light loads.

Artificially modified layers

Phosphating. The protection afforded by the normal type of regenerative film, although generally sufficient for most bearing conditions, may not be adequate when extreme pressures are developed between the sliding surfaces. In such cases additional protection may usually be obtained by artificially modifying the surface layers of the rubbing surfaces so that sliding takes place in effect between two non-metallic surfaces.

In the case of ferrous specimens, one of the most successful processes that has been used is that in which a deep phosphate coating is formed on the surface of the specimen. Because of the porous structure of such a coating, it tends to hold the lubricant. Moreover, owing to the method of formation of the coating, it is strongly bound to the underlying ferrous substrata. It is comparatively friable and rapidly forms a smooth bearing surface, so that hydrodynamic conditions are established almost immediately. As an indication of its anti-seizure properties, tests carried out at Thorntonhall with a four-ball machine showed that in the case of one particular lubricant the seizure load was increased from 70 to 700 kg when the balls were treated in this manner.

Barwell and Milne[14] have shown that the effectiveness of a bonded molybdenum disulphide film can be greatly increased when used in conjunction with a phosphate treatment. Surfaces of this nature will withstand considerable loads for prolonged periods without seizure or appreciable wear. Only a comparatively poor bearing surface is obtained if molybdenum disulphide is rubbed into the surface, and it appears necessary to bond the material firmly to the phosphate coating if it is to be successful. Although this latter procedure involves heating the surface to 350 deg C, this is not necessarily such a severe limitation as it first appears, since the use of such a dry lubricant is particularly suitable when temperature considerations rule out the employment of conventional lubricating oils. Some alternative method of application, however, would be desirable, but is not available at present.

Phosphating has also proved to be an extremely valuable palliative for fretting corrosion. Owing to the restricted motion of the two surfaces, the rubbing conditions can be severe, and it is therefore not possible for hydrodynamic films to be established. The application of a phosphate film, however, not only prevents metallic contact, but also enables an oil film to be maintained, and so aids in reducing the oxidizing conditions at the rubbing surfaces.

Anodized surfaces. Particular interest has recently been shown in the possibilities of using anodized surfaces as a method of overcoming the extremely poor bearing characteristics of aluminium. Two factors point to their possible utility:

1. They have a very porous structure, and therefore are ideally suited for either dry or oil lubrication.
2. They possess a considerable rigidity, though owing to the low yield strength of the underlying material it is necessary to form as thick a coating as possible.

One major disadvantage of anodized surfaces is their very poor thermal conductivity. This restricts their application to bearings in which running conditions are fairly light. Under severe conditions, failure to remove the frictional heat leads to a rapid distintegration of the film, and subsequent seizure of the bearing surfaces. Some alleviation from this restriction, however, may be obtained by using such a surface in conjunction with a mating material of good thermal conductivity, and by removing as much heat as possible by the use of an adequate flow of lubricant.

Experimental results obtained at Thorntonhall indicate that the frictional force between anodized surfaces may be due primarily to the interlocking of asperities, and is an interesting example of the form of friction suggested by Bikerman[15]. The frictional characteristics of the very smooth anodized surfaces are not greatly affected by the use of either B.P. paraffin or oleic acid as lubricants. With surfaces the average roughness of

which is 40 micro-in, the addition of a fatty acid as lubricant reduces the friction coefficient to the low value of 0·15, whereas B.P. paraffin has little or no effect. This seems to indicate either that soap formation occurs, or that in the presence of the surface-active liquid the asperities are weakened by its penetration into the fine porous structure of the film.

Surface layer reaction with a lubricant

A great deal of work has been published on this subject, and it is not possible to give here a complete review of the discoveries made in this field. It may be useful, however, to enumerate the various factors involved, and to mention those aspects of the problem which have received little attention in the past.

The object of forming a protective surface by chemical reaction is to produce a film of relatively low shear strength to facilitate sliding. The process by means of which the protective film is being formed depends on the following factors:

1. The reactivity of the metal surface.
2. The nature of the lubricant present.
3. The mechanical conditions of sliding, such as speed and load, which determine the temperature and pressure at the surfaces.
4. The conditions of environment, for example the presence of oxygen and water.

There is great interaction between the various factors, and a change in any one of these alters the properties of the protective film on the surfaces.

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PROOF TESTING OF COMPONENTS

IN an article by A. L. Boegehold in *Metal Progress*, 1952, Vol. 62, the proof of testing of full size automobile components is discussed. It is stated that to translate the results of laboratory strength tests on simple test pieces into safe design figures for machine parts operating under dynamic loading is a difficult matter.

After making laboratory fatigue tests on specimens of different steels heat treated in various ways, these steels were used to construct rear axle ring gears which were assembled in a complete rear axle. Fatigue testing was then performed in a machine in which the propeller shaft is driven by a dynamometer motor, and at the ends of the axle, the wheels are replaced by electric generators in which the field strength is variable so that the load on the rear axle may be controlled.

Contradictory results were obtained from the two sets of fatigue tests. Reasons for the discrepancies are discussed, and it is explained that had enough experiments with the different steels been made to determine the way to cut the teeth in order to get the best pattern of tooth contact, the results obtained in the rear axle tests would have been entirely different, and this was confirmed in subsequent work.

A special apparatus for testing steel in crankshafts was used to examine a section of the crankshaft comprising two main bearings, one crank throw, and the crankpin. A small motor with an eccentric weight on the armature shaft is hung on one of two heavy weights clamped to the main journal bearings. As it rotates the motor causes the weight to vibrate. The motion is transmitted through the crankshaft to the other weight, so that a resonant vibration is set up at the natural frequency of the whole system. The stress in the crankshaft is controlled by a contactor in an electronic circuit which regulates the amplitude of vibration as a function of exciter motor speed.

General purpose machines for testing automobile components include the Sonntag fatigue tester, a constant-load machine, the General Motors fatigue testing machine developed by Underwood and his associates, the Chevrolet dual-purpose stroking machine, which may be operated either as a constant-stroke or a constant-load machine, and a machine using a similar means for stroking, originally developed by Almen. The use of testing machines is described in an investigation into the failure of a pitman shaft in a bus that left the road and hit a concrete pier. (*M.I.R.A. Abstract No.* 6033).

THE HYDRAULIC TORQUE CONVERTER

A PAPER by R. M. Schaefer and J. A. Winter, *S.A.E. Preprint*, September 9-11, 1952, describes an investigation into the relative transmission stresses encountered in trucks equipped with mechanical and hydraulic torque-converter drives respectively. The test programme was carried out on a number of trucks engaged in "off-the-highway" earthmoving, and coal and ore hauling. Trucks of each type were equipped with a torquemeter on the propeller shaft, comprising a strain-gauge network connected to a Burroughs mercury-bath commutator, or pick-up, together with a suitable amplifying and recording instrument.

Torque recordings were taken during starting, gear changing of mechanical units, and over extensive routes, on trucks equipped with both mechanical and torque-converter drives, but otherwise identical. The recordings are illustrated and compared, and the sharp positive and negative torque peaks which occur with the mechanical drive are clearly shown.

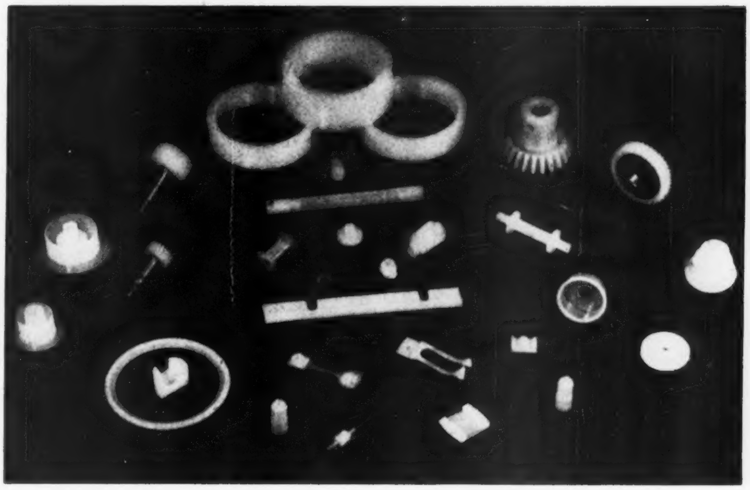
During starting with heavy loads, the clutch engagement shock load was approximately 4¼ times the converter observed starting torque, the clutch shock loads including serious torque reversals. Shock loads during gear changing exceeded the smoothly applied torque of the converter by about the same ratio. Reactions due to torque variation under normal operation varied by ±28 per cent about the average mean load with the mechanical unit, but by only ±14 per cent with the converter. It was found in general that design stresses for converter-driven vehicles could be based upon an engine torque 1·25 times that used for an equivalent mechanically driven vehicle, to give approximately the same life expectancy.

The effect on fatigue life of operating under varying stress is discussed, and it is shown that the torque-converter drive permits a higher load factor, a longer life factor, and reduced operating costs to the user. (*M.I.R.A. Abstract No.* 6075.)

MOULDED NYLON

A Plastic Development with Great Potentialities

NYLON gears, nylon bearings, nylon washers, and a great variety of other moulded nylon parts are now being used in large quantities by textile machinery makers, by the automobile industry, and by other manufacturers who, after prolonged field tests, have been convinced of nylon's superior qualities and proportionately low price/life ratio. As a textile fibre, nylon is well known. Hosiery, industrial cloths, sheer dress fabrics, lingerie fabrics, and ropes, have all helped to make nylon a household word throughout the world. Now it is making its mark as a moulded plastic, competing not only with other plastics, but with steel, brass, copper and other metals.

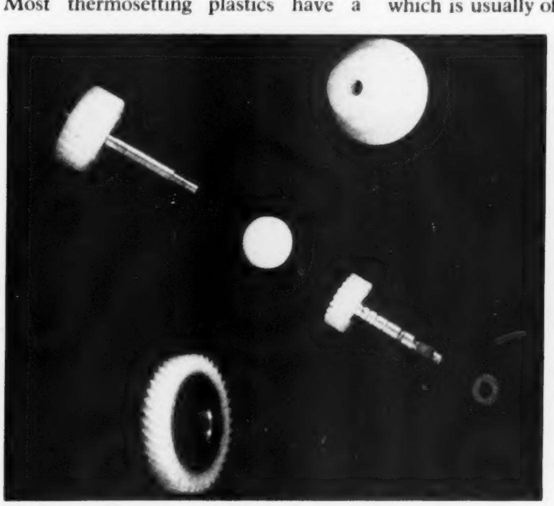


Machinery parts moulded from nylon by British Ropes Ltd.

Moulded nylon has been available in the United States for several years and to-day there are hundreds of applications for nylon parts and components are produced on a large scale. Owing to the limited production capacity of the polymer manufacturers in this country, progress has been less rapid. However, a steady improvement is taking place and great advances are now being made with the introduction of nylon as a material for gears, bearings, textile machine parts, etc. British Ropes Limited—already well known for their pioneering of the use of nylon for rope-making; and for adapting nylon cloth to industrial applications such as filters and anode bags—are now manufacturing nylon mouldings in their Plastics Department at Leith, Edinburgh.

Nylon is a thermoplastic material and may be re-heated and re-shaped many times although some of the physical properties are adversely affected if the re-cycling is carried out too frequently. Most thermosetting plastics have a service temperature of somewhere in the region of 180°F, but nylon does not soften until the temperature rises above 300°F, and it melts at temperatures above 480°, which is generally higher than the charring point of most thermosetting materials.

Nylon parts are produced by the injection moulding process which consists, basically, in the application of heat to the moulding powder in an electrically heated cylinder until the material reaches melting point. At this stage it assumes the form of a viscous syrup-like liquid, which is then forced under considerable pressure (up to 20,000 p.s.i.) into a precision mould which is usually of the highest grade of tool steel. On contact with the cold mould, the plastic nylon chills and resumes its solid form and assumes the shape of the mould cavities. After a brief cooling period the mould is automatically separated and the finished products removed. The whole moulding cycle is generally completely automatic and the overall cycle may be as short as 45 seconds, but this is dependent on the section being produced.



Moulded nylon parts for automobile and general machinery applications

This process is particularly suitable for the mass production of parts in bulk, and providing the quantities to be produced justify the initial mould cost, the moulded nylon parts compare very favourably in price with metal items which require individual machining operations. This is particularly noticeable in the case of moulded nylon gears. If the gear shape is a complex one a great saving is made by moulding, as the intricacy of form does not add to the moulding cost as it does to the machining cost.

Nylon has many outstanding physical and mechanical properties, the most important being:

- Extreme toughness (especially at low temperatures).
- Abrasion resistance.
- Form stability at high temperatures.
- Strength in extremely thin sections.
- Lightness in weight (specific gravity = 1·14).
- Chemical resistance (except to strong acids).
- Non-inflammability.
- Low co-efficient of friction.

The fact that nylon can be used at high speeds under light loads without lubrication has led to its wide use in the textile industry with considerable saving through elimination of damage to goods by oil contamination. For this same reason, nylon has found favour in the food handling and packaging industry. However, it should be understood that, as with any other bearing material, the generation of heat can be reduced by lubrication, and the effectiveness of the lubricant will determine to a large degree the temperature which will be reached under certain conditions of speed and load on any particular piece of equipment.

Obviously, then, nylon should be lubricated when possible, but it is a virtue of nylon that it will operate more satisfactorily than other bearing materials under conditions of imperfect lubrication or even with no lubrication at all.

The following figures, showing the comparison of the weight per cubic foot of nylon with a few of the more common metals, indicate that weight can be considerably reduced by changing over to moulded nylon.

	lb/cu ft.
Nylon	71
Laminated plastics	84
Aluminium	169
Copper	530
Brass	535
Bronze	548
Zinc	417
Steel	488
Cast iron	455

From experience gained to date on nylon gears, it would seem that the ability of nylon to damp mechanical vibration in the audible spectrum will reduce to a minimum the noise in a train of gears. This is particularly desirable in motion picture and sound recording apparatus and is a great advantage in all types of household equipment such as mixing and washing machines.

In general it has been found that the rate of wear on nylon gears is greatest at the start and decreases with further operation. The rate of wear seems to reach a minimum after about 0·001/0·002 in has been removed from the surface of the nylon and it seems probable that in contact with steel and brass this is indicative of the amount of nylon which must be removed in the course of smoothing the metal sufficiently so that it does not cause further wear of the nylon.

In most cases the co-efficient of friction decreases from the start up to about four hours and then remains constant—after this time the rate of wear frequently becomes negligible with a consequent unusually long service life. Nylon to nylon seems to give the best performance in gear trains, with nylon to steel almost as good and nylon to brass showing a poorer result.

It is possible to machine moulded nylon in much the same way as other thermoplastics. Dissipation of heat is an important feature and the rate at which frictional heat can be reduced is usually the controlling factor in determining the speed at which machining can be performed. The ability of nylon to be machined often solves the problem of small quantity order or prototype experimentation, as blanks can be moulded to near-size from which the finished parts can be produced.

With all the properties mentioned heretofore, it is not surprising that the field for nylon mouldings is considered unlimited and it would be quite impossible to list all the potential uses, but it is opportune to mention a few of the proved applications as a guide to the potential user.

Lock-nut inserts from nylon have largely replaced the fibre inserts previously used.

Speedometer gears. A five-step operation formerly employed in machining steel gears has now been replaced with a single operation in which the gear is moulded from nylon directly on to a steel shaft. Bend tests have shown that the nylon gears can withstand 100,000 miles of operation at 80 m.p.h. without visible signs of wear.

Coil formers of nylon are finding more and more favour in the electrical industry for many reasons. The ability to mould nylon into thin sections whilst still retaining a high degree of strength reduces both bulk and weight. The excellent heat resistance of nylon enables terminal blocks or lugs to be moulded into the former; this tends to eliminate breakage of wire leads in service.

There is little doubt that there is an application for moulded nylon in practically every industry and the Plastics Department of British Ropes Limited, Leith, Edinburgh, who have specialized in the moulding of nylon for the past three years, are always ready to investigate new uses and to give further technical information to prospective users.

NEW EXIDE BATTERIES

AN entirely new range of heavy duty batteries for commercial vehicles has just been introduced by Chloride Batteries Ltd., Manchester. It is known as the Exide KHV series and is available in capacities ranging from 81 amp-hr to 135 amp-hr in the 6 volt, and 67 amp-hr to 135 amp-hr in the 12 volt sizes.

The manufacturers claim that these new batteries have a starting performance of as much as 35 per cent better than that of many equivalent types commonly in use, they are less bulky, and up to a 20 per cent saving in weight by comparison with similar, more conventional units has been effected. They further state that exhaustive trials have shown that the KHV batteries will set new standards in trouble-free operation and working life, particularly under adverse conditions. All these characteristics, by giving faster engine cranking speeds and a greater reserve of power, will ensure quicker starting at low temperatures. This will be a great advantage to diesel operators.

Furthermore, it will be possible to fit a heavy duty battery on those light commercial vehicles which hitherto have had to rely upon a car type. This will give the same capacity and starting performance as before, but from a battery with greater robustness and longer life. The high electrical efficiency and low weight and bulk of these batteries are atttained by the provision of a larger number of plates designed to work in conjunction with a new system of dual separation, comprising porvic plastics separators and glass wool retainers. Continuous research in the Exide laboratories has resulted not only in the development, for plate grids, of the new lead alloy, CB.95, with a greater mechanical strength and resistance to corrosion, but also in improved active materials which give the new batteries better performance.

Exide KHV battery with dual separation system

Porvic as a separating material offers outstanding advantages over other types. Conventional separators are attacked chemically by the positive active material, causing excessive wear, and a resultant loosening of the internal assembly. This in turn allows the positive plates to buckle, thus leading to internal short circuits and premature battery failure. Besides being strong mechanically, Porvic is also immune to oxidation and acid attack even under conditions of severe over-charging; and a tight assembly is therefore maintained throughout the life of the battery. Its porosity is over 80 per cent, with the result that the acid is freely absorbed by the new separators so that the battery may develop its full power instantaneously to supply the starter motor. The electrical resistance of this material is as low as that of the highest quality conventional separators. The manufacturers state that Porvic can withstand without deterioration higher temperatures than any other separator material, in fact it remains unaffected by temperatures which would ruin the plates themselves.

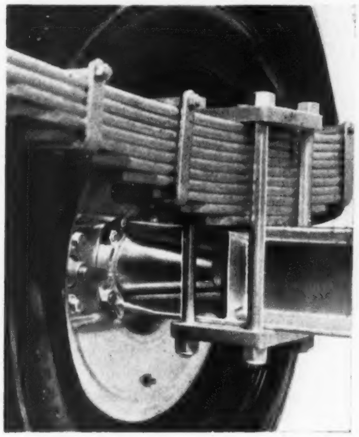
TIPPERS AND TRAILERS

Recent Interesting Developments

As may often be seen at a Commercial Vehicle Show, partly on account of small-scale manufacture and partly as a consequence of the widely varying purposes for which they are used, more ingenuity is shown in trailers than in self-propelled vehicles. The rear axle of the Hands trailer is a good example. It has an ordinary rolled joist section as its beam and the two stubs are made with conical ends which are fitted into corresponding vee-shapes, flame-cut at each end of the web of the beam, and united by fillet welds on each side. The spring seats directly on the top flange of the joist, which is reinforced by vertical distance pieces welded immediately below it.

Another ingenious detail of the Hands trailer is the compensating mechanism for the rear brakes. These are operated from a flexible cable running centrally down the chassis and then curving down to one side to take an abutment inside a stirrup welded to a rod coupled to the pull-on expander of the Girling brake on one end of the axle. The inner member of the cable itself is attached to the other expander and on application the reaction on the abutment applies one brake while the pull of the cable works the other end and accurate compensation is automatically obtained.

Taskers of Andover produce an interesting semi-trailer with detachable rear bogie. This comprises a massive cross bar on which oscillate two box-section members carrying the stubs on the front and rear wheels on each side. The end of the frame hooks slightly over the cross bar and is held in position by two massive U-bolts embracing curved horns welded to the frame just ahead of the cross bar. It is only necessary to slack off the nuts on these a few turns until they and their clamping plate can be moved clear of the horns when the frame can be jacked up clear of the cross bar and afterwards lowered to remove the load.



Axle stub welded to I-beam on Hands trailer axle

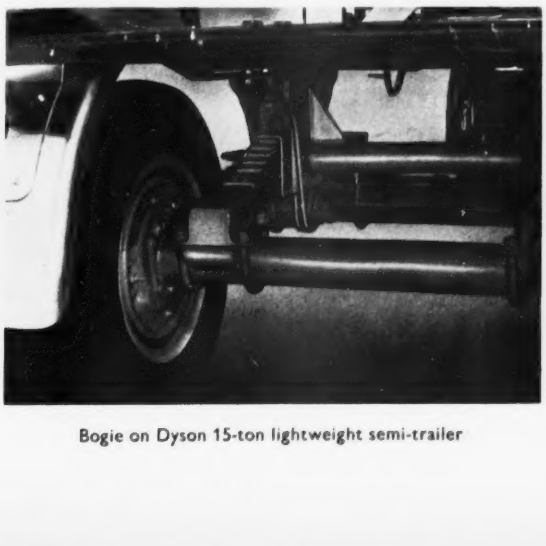
Another neat, but not detachable, rear bogie is to be found on a Hands trailer in which the two axles are connected by inverted semi-elliptic springs. The central spring box oscillates between two vertical webs of a welded bracket and since any spring bolts between these webs would be very inaccessible for taking up any slackness, the spring is actualy retained in a completely embracing buckle with two opposing wedges on top drawn together by a bolt whose nut can easily be got at with a box spanner from the back. We note that in this instance all braking strains are taken by the ends of the leaf springs which are enclosed in a box containing a rubber mounting.

Among interesting designs of trailer axle is one having a load rating of 18,000 lb made by Rubery Owen and Co., Ltd. This followed the design employed for so many years on the Scammell mechanical horse in which the axle is made up of a Mannesmann tube with up-set flanges bolted to up-set stubs by a large number of small bolts which also secure the brake backplates. In the case of the Scammell axles the spring seats are castings of weldable malleable iron and require no machining whatever before welding to the tube. Care is taken to interrupt the weld in the region of the top of the axle, this part being left to avoid overheating a highly stressed part of the tube.

Tipping bodies for rough work

Until recent years most tippers built in this country were operated under reasonably easy conditions; a normal chassis frame with slight local stiffening and telescopic ram or pair of rams at the body would work satisfactorily. Very large tippers are now being built for constructional work in this country and abroad, and operated under the most brutally severe conditions. As they may have to tip their full load with one of the back wheels sunk in soft ground, the design of tipping bodies and their attachments has received a great deal of consideration. The trouble is, of course, that the lateral displacement of the body and twisting of the chassis frame throws strains on the telescopic rams which they were never intended to take. Especially in the case of the large tippers, such as those made by Scammell Lorries Ltd., the bodies themselves are enormously stiff in torsion. The main tank, which often has a slightly upturned rear instead of a door, is reinforced by large mem-

Detachable bogie on Taskers 25-ton semi-trailer



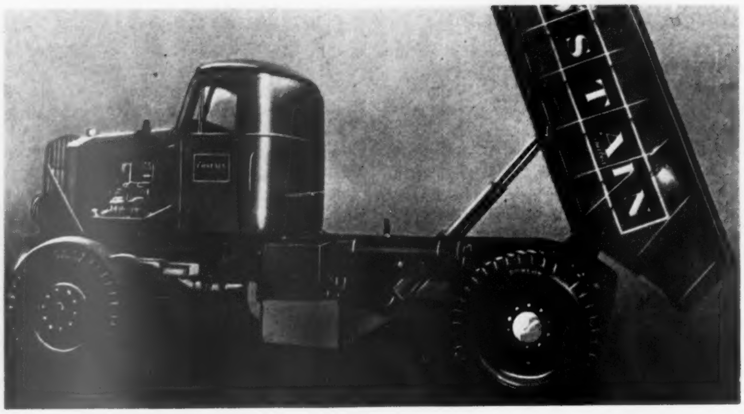
Bogie on Dyson 15-ton lightweight semi-trailer

bers of channel section pressings wrapped round it with the tips of their flanges welded to the body shell.

The vital point as far as the vehicle frame is concerned is to provide the maximum torsional stiffness between the body pivots at the rear and the region of the attachment of the rear springs and the axle, or axles. It is quite unnecessary to provide torsional stiffness ahead of this point because when tipping a load there is hardly any weight on the front springs and they make no worth-while contribution to the stability of the vehicle.

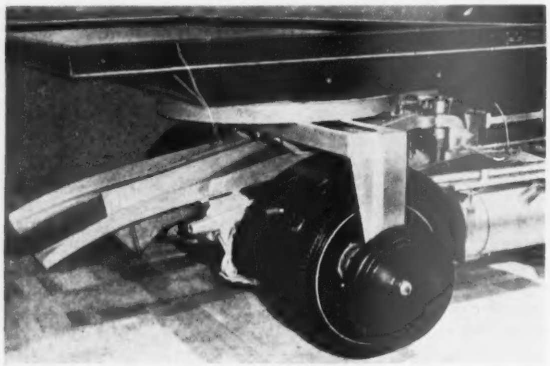
On a body made by the Telehoist concern the problem has been attacked by laterally stabilizing the body through a long triangulated stay with its two lower corners hinged to the chassis at a point well in front of the back axle and its apex pin-jointed to a slipper running up and down a rail welded to the bottom of the body.

A similar effect is produced on the Bedford tipper by a stay consisting of two rectangular assemblies hinged together. Each rectangle is built up of steel pressings with cross bracing, which makes them torsionally very stiff. This is a point of some importance. When not quite fully extended, the lower member is almost horizontal while the upper one is nearly vertical, and the arrangement can only give lateral stability if the lower half is really stiff in torsion.

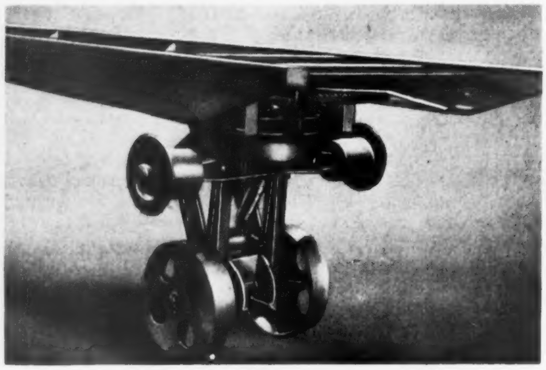


Scammell tipper

On the Scammell Mountaineer tipper, the rear of the frame is given great torsional stiffness by the incorporation of an enormous tubular cross-member. The telescopic Telehoist rams are ball jointed to brackets on this member and on the body so that, whatever happens, no bending strain is imposed on the ram assembly. Incidentally, the Scammell Mountaineer has a body without a rear door and has, therefore, to be tipped to an angle of about 75 deg to discharge its load. In that position the centre of gravity of the body, which includes a massive canopy to protect the driver's cab, is well behind the tipping pivots. Towards the end of the tipping motion the body picks up a hanging lever attached to an hydraulic cylinder. This serves first as a cushion to check the last part of movement of the body and, secondly, when the control valve is changed over, operates to return the body by hydraulic power.



Coupling gear on Karrier mechanical horse

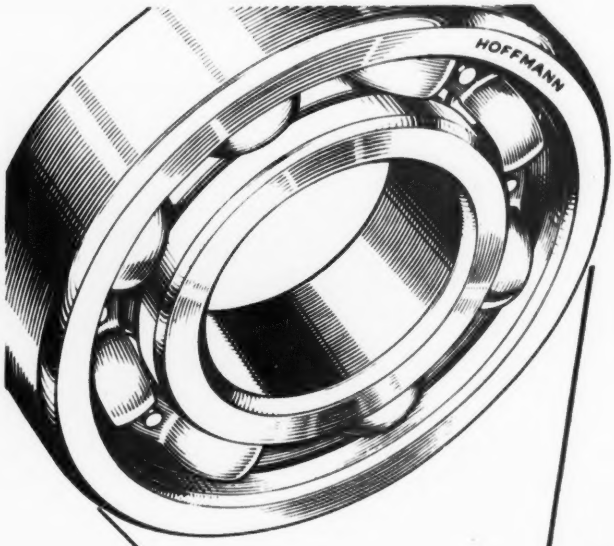


Commer-Hands 8-ton semi-trailer chassis

ENGINE—TRANSMISSION RELATIONSHIP

IN an *S.A.E. Preprint*, June 1-6, 1952, D. F. Caris and R. A. Richardson claim that very substantial gains in fuel economy can be achieved by the use of an ideal engine-transmission relationship. This is one in which the transmission will at all times keep the engine operating at that speed and throttle opening which give minimum specific fuel consumption for the particular horse-power required at the time. A family of fuel consumption curves, described as cross-section curves, showing specific fuel consumption plotted to a base of horse-power for various engine speeds, shows how an envelope curve can be drawn round the points of minimum consumption. With an ideal transmission, the engine will always operate at some point on this envelope and thus will seldom operate at a power factor much below 70-80 per cent. The poor specific fuel consumptions normally associated with road-load running at low throttle openings will thus be eliminated.

In a normal engine, working with the ideal transmission, it is claimed that the operation at high load factor will eliminate plug fouling and, to a large extent, combustion chamber deposits, thus adding "mechanical octane numbers," while the vacuum spark advance can probably be eliminated. Careful attention to valve timing is required to ensure good high-speed breathing together with solid low-speed performance. If high compression ratios are to be used, it will be found that main bearings on each side of each crankpin are required for smoothness, that overhead valves are essential, and that a low stroke : bore ratio is desirable. It is shown that variation of stroke : bore ratio within the limits 0·71 to 1·33 does not affect economy. (*M.I.R.A. Abstract No.* 5952.)



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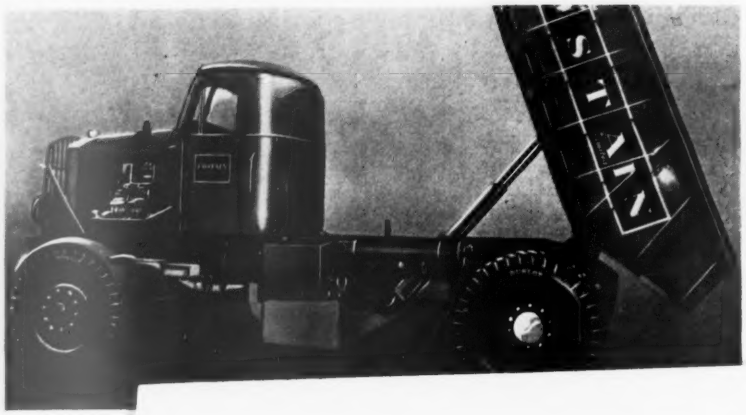
On a body made by the Telehoist concern the problem has been attacked by laterally stabilizing the body through a long triangulated stay with its two lower corners hinged to the chassis at a point well in front of the back axle and its apex pin-jointed to a slipper running up and down a rail welded to the bottom of the body.

A similar effect is produced on the Bedford tipper by a stay consisting of two rectangular assemblies hinged together. Each rectangle is built up of steel pressings with cross bracing, which makes them torsionally very stiff. This is a point of some importance. When not quite fully extended, the lower member is almost horizontal while the upper one is nearly vertical, and the arrangement can only give lateral stability if the lower half is



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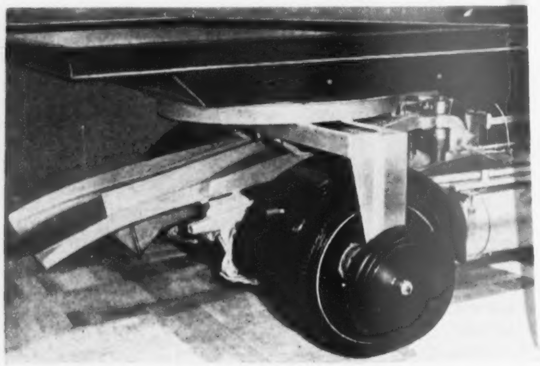

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ENGINE—TRANSMISSION RELATIONSHIP

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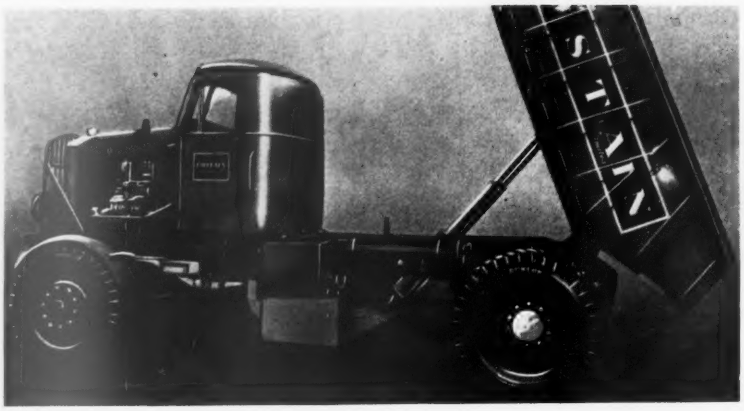

Scammell tipper

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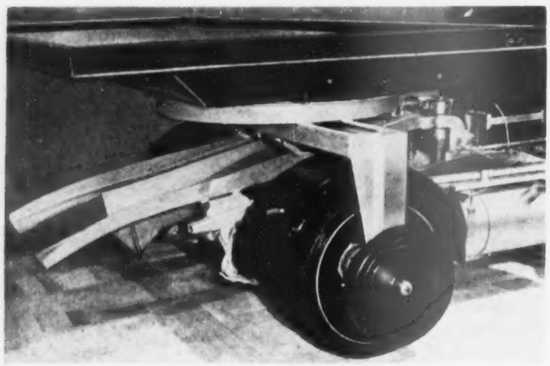

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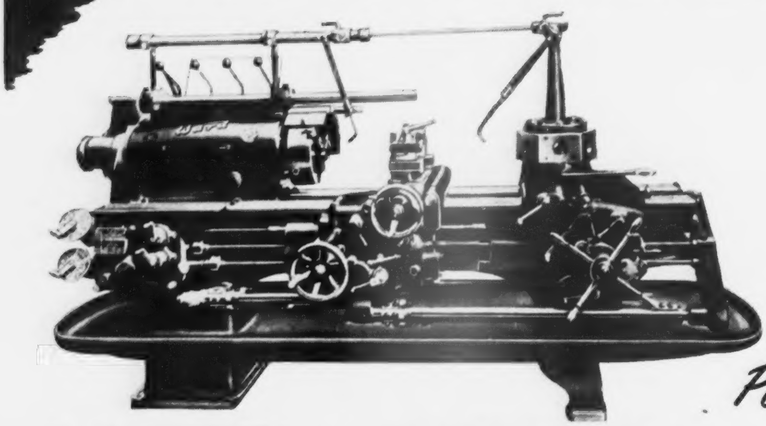

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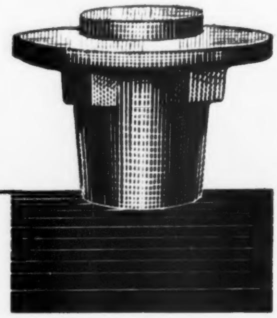

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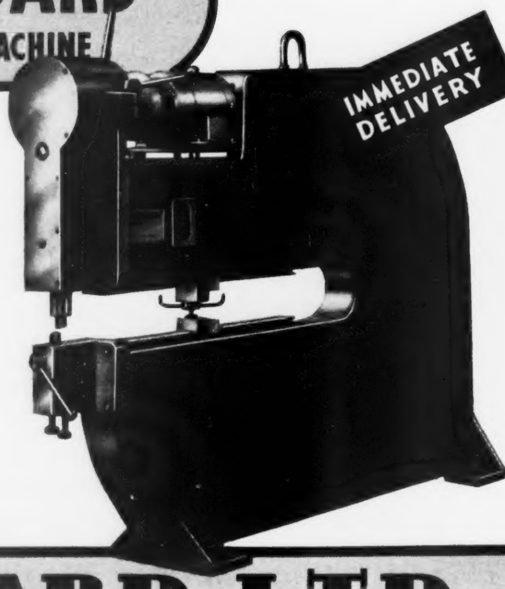

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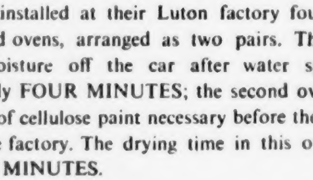

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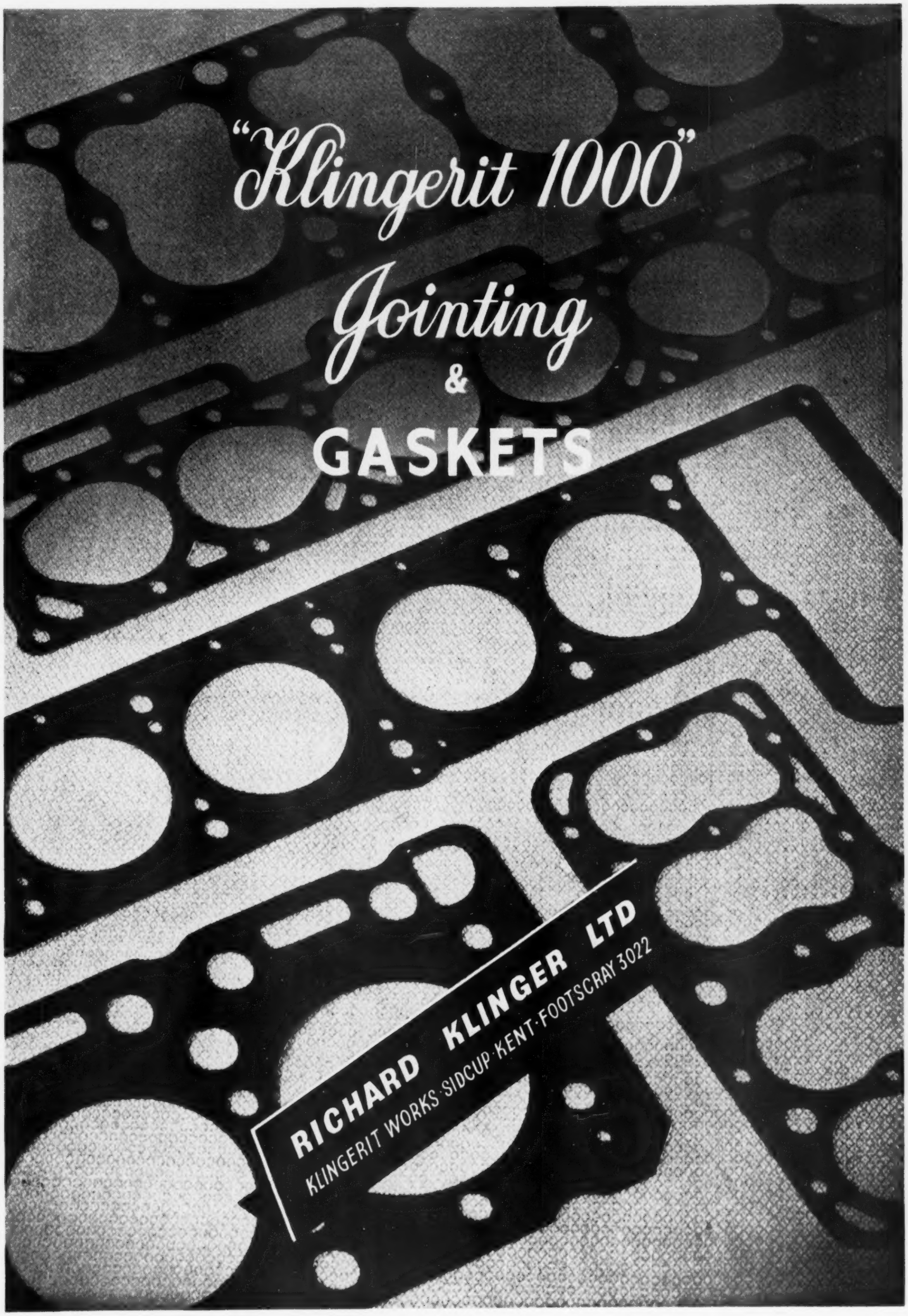
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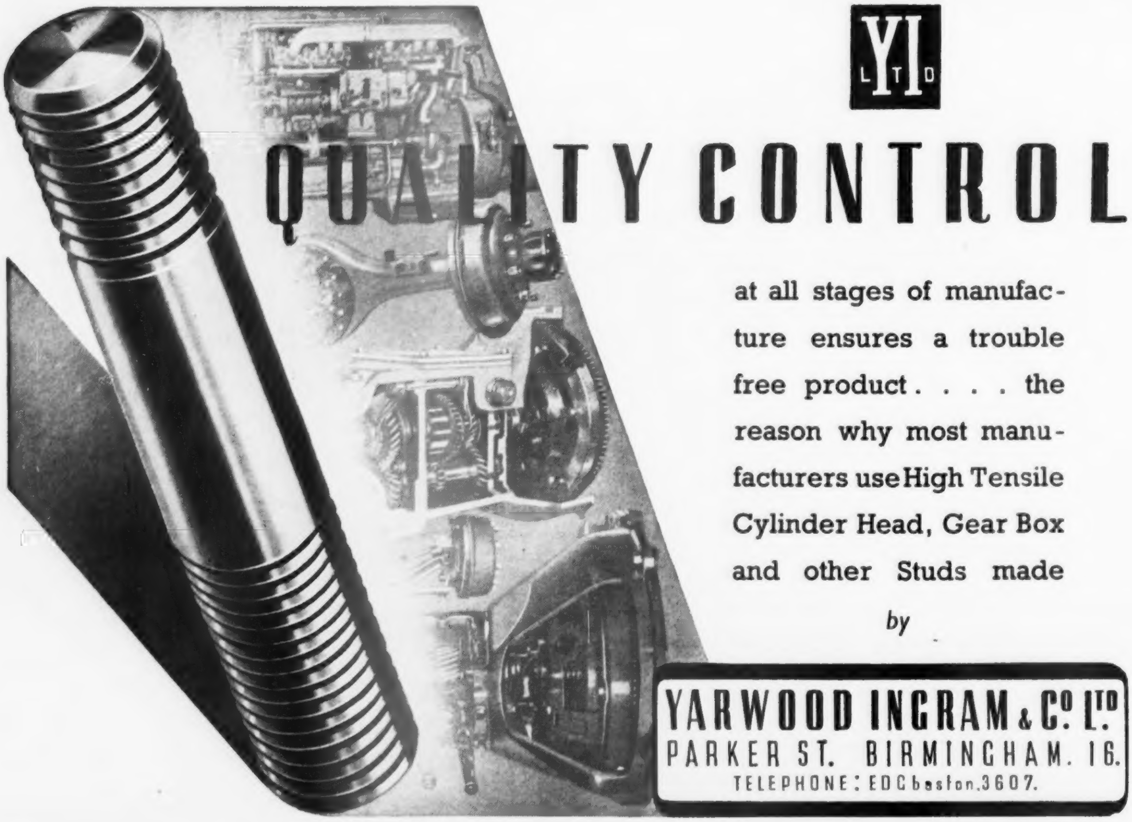
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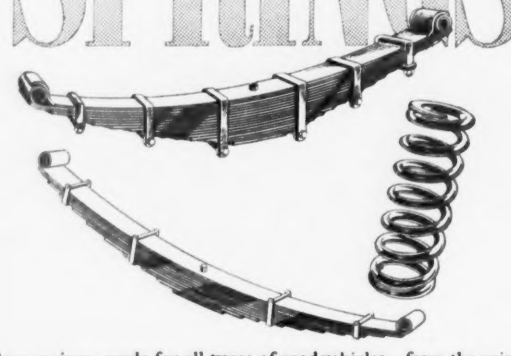

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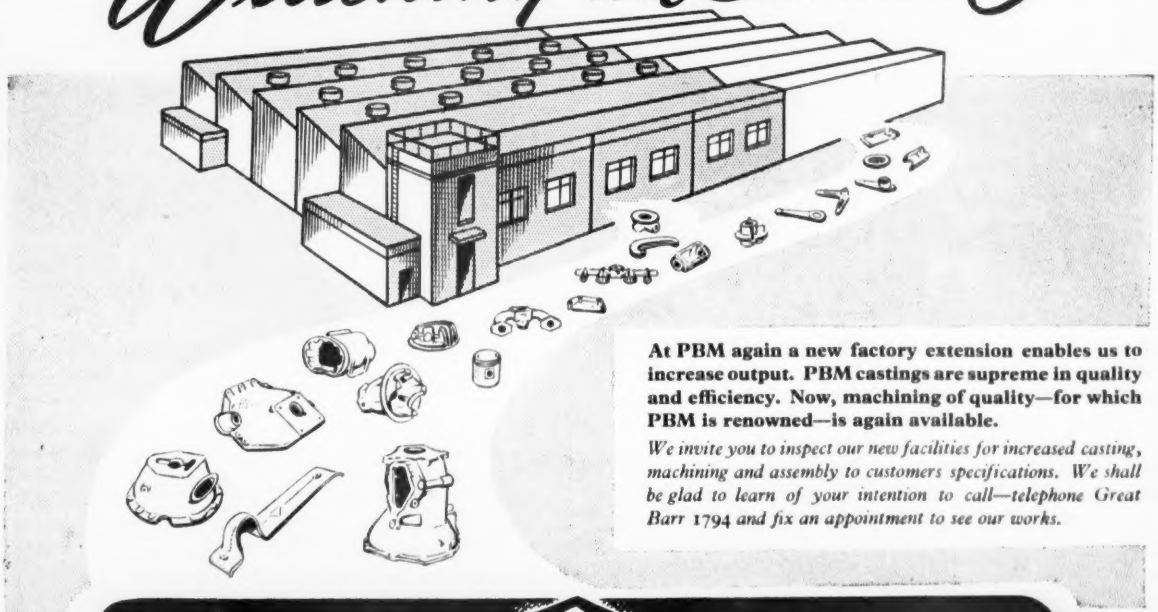

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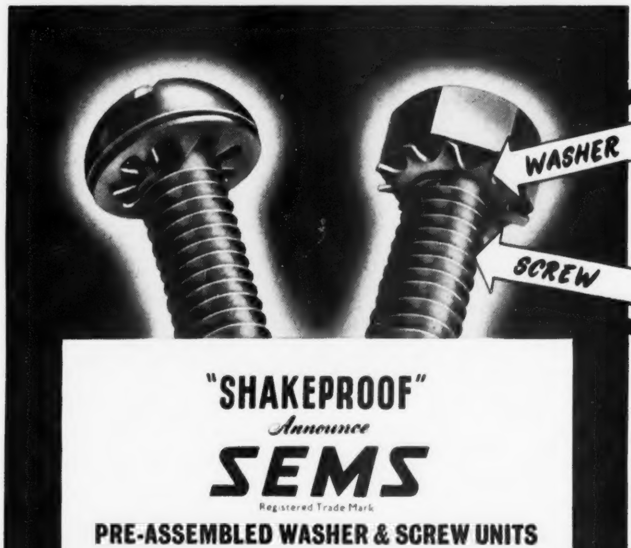

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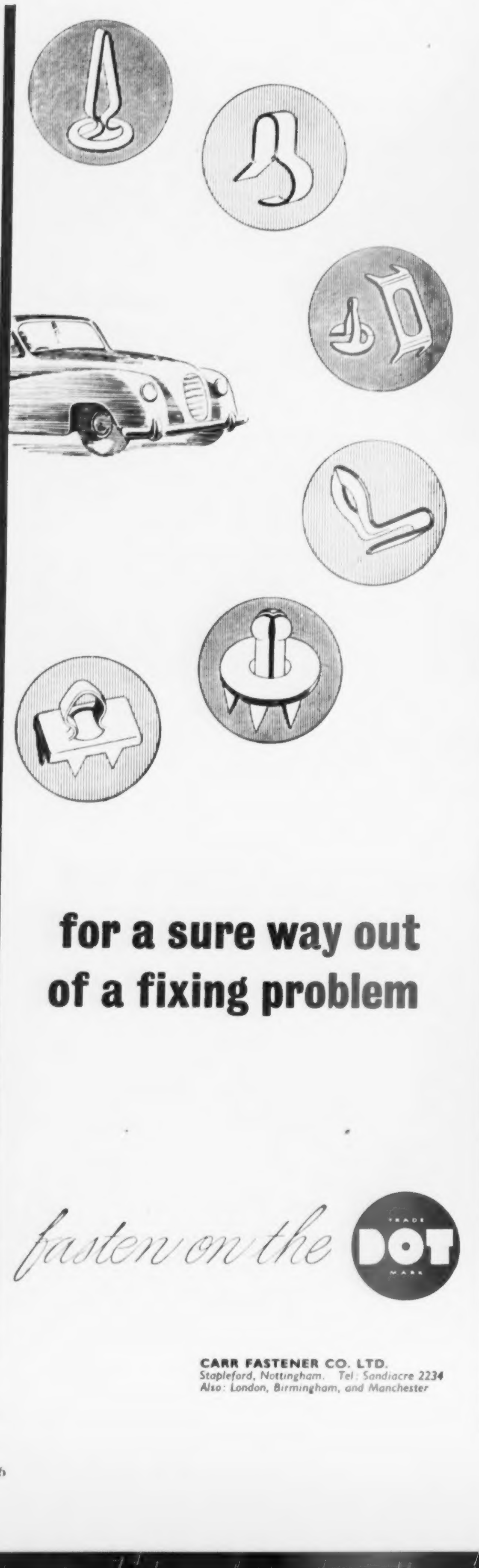
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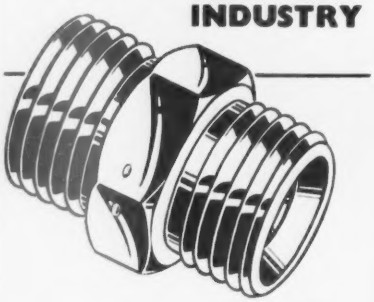

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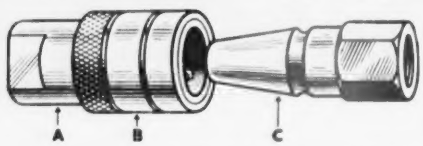

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